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DEVELOPMENT OF HELICOPTER TRANSMISSION SEALS

TASK II

BY

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Sikorsky Aircraft

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16. Abstract

Two high speed helicopter transmission seal concepts were designed, fabricated and tested. The concepts were a dual element split ring seal and a circumferential seal. The tests were performed in a rig using an actual input quill assembly. The test conditions were selected to simulate transmission operation and were 383 K (230°F) oil temperature, and a sliding speed of 2865 m/min. (9400 ft/min). The split ring seal exhibited gross leakage and was considered unsatisfactory, while the circumferential seal leakage was less than 1 c.c./hour; this leakage is within acceptable limits. The circumferential seal wear was only 0 to 0.0127 mm (0 to .0005 inches) during a 100 hour run (40 starts and stops). During a 40 hour contamination test (mesh silica flour) the seal total wear was a maximum of .1 mm (.004 inches); this wear is considered acceptable.

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FOREWORD

The report describes the work conducted during Task II of Contract NAS 3-15684 by Sikorsky Aircraft Division of United Aircraft Corporation, Stratford, Connecticut. Task II was initiated on 22 June 1971 and completed on 21 June 1972. The work was funded by the U.S. Army Air Mobility Research and Development Laboratory, Moffett Field, California; Technical management was performed by personnel of the Lewis Research Center of the National Aeronautics and Space Administration. Mr. L. P. Ludwig, Fluid System Components Division, NASA Lewis Research Center was the Project Manager. Mr. L. W. Schopen, NASA Lewis Research Center was the Contracting Officer.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF FIGURES

LIST OF TABLES

SUMMARY

From a screening study of nine (9) different seal configurations, two configurations judged potentially useful in a helicopter high speed input shaft application were selected for design, fabrication and test. The seals selected were a dual element split ring seal and a circumferential seal.

The seals were evaluated at 2865 m/min (9400 ft/min), 383 K (230°F) oil temperature and with oil flows that duplicated flow rates of actual operation. Seal pressure differentials was determined by these oil flow rates and scavenge capacity and was near zero. The test rig incorporated an actual transmission input quill assembly. The test was divided into a 25-hour screening test and a 100-hour endurance test.

The dual element split ring seal, a controlled clearance seal, could not be operated successfully. Excessive leakage occurred with the production drainage configuration and with modifications for increased drainage. Although not an acceptable solution, a leakage rate of 2 c.c./hr. was obtained with an auxiliary pump scavenging lubricant from the seal cavity. It was concluded that a viable approach using a dual element split ring seal could not be obtained without excellent lubricant drainage in the seal region.

The circumferential seal, a conventional segmented three ring design, exhibited leakage of 12 c.c./hr. during the initial screening test. The seal was reworked by doubling the radial garter spring load. The reworked seal was then successfully operated for 25 hours with an average leakage rate of .74 c.c./hour.

Another circumferential seal was reworked to conform to the same design parameters of the first successful seal. It was operated for 100 hours with an average leakage rate of .56 c.c./hour. The test was performed with a start/stop cycle of two hours. Maximum wear of the seal inside diameter was .0127 m.m. (.0005 in.), while average wear was .00508 m.m. (.0002 in.).

A 40-hour environmental test, simulating exposure to South-east Asia dust, was also conducted by subjecting the seal assembly to periodic injection of 140 mesh silica flour. During 16 hours of this test, oil flow to the seal was increased to determine the affect of flooding on seal performance. Leakage rates of .124 c.c./hr. without flooding and .095 c.c./hr. with flooding were obtained. Maximum wear of the wear sleeve outside diameter was .071 mm (.0028 inches) and wear of the seal inside diameter was .0305 mm (.0012 inches).

Introduction

Helicopter transmission high speed oil seals have been a chronic problem often resulting in premature gearbox removal due to excessive leakage. Most of these high speed seals are located in the engine input section of the gearbox. The intent of Contract NAS3-15684 is to develop high speed seals for current and future transmission applications.

The function of these seals is to prevent lubricant leakage and to prevent dirt, water and debris from getting into the transmission. Seal failure is not usually catastrophic, but can be a significant cost factor if failure is frequent. The cost factor involves not only replacement cost but aircraft down time; also exposure of the system to environment during assembly might allow introduction of debris or dirt. Further, the probability of misassembly is present. Therefore it is highly desirable to have the seals operate at least for the time between transmission overhauls. Helicopter transmissions operate with low pressure differentials across the seal - probably not more than .689 N/cm^2 (1 psi) in most cases. The seals usually operate at speeds of 152 m/min (500 ft/min) to 4550 m/min (15000 ft/min). The sealed media is usually an ester base synthetic engine oil (MIL-L-23699, MIL-L-7808) or a mineral base oil (MIL-L-21260) at a temperature of 366 K to 394 K (200 to 250°F). Currently most helicopter transmission high speed seals are face seals, Figure 1; circumferential seals, Figure 2; or radial lip seals, Figure 3.

These seals fall into the general classification of positive-contact seals. A full or partial fluid film exists between the sliding sealing surfaces. Leakage of these seals in successful applications is less than 2cc/hour.

Elastomeric type lip seals are the most commonly used seal for helicopters. However, the elastomer material in this type of seal has limited abrasion resistance (dirt) and limited temperature capability. At high sliding speeds the temperature of the lubricating film between the elastomeric lip and the shaft is much higher than the bulk lubricant temperature. Thus, in effect, the temperature rise in the lubricating film determines the useful maximum speed of elastomeric lip seals. And evidence (ref. 1) is that oil film temperature in thin films under shear can reach 533 K (500°F) at the relatively low sliding speeds of 610 m/min (2000 ft/min). It follows that high speed seals should have materials that can withstand the high temperatures of the lubricating film.

The objectives of this study were to: (a) review current seal technology and select two candidate seals judged suitable for high speed operation, (b) build and test the two candidate seals that have axial lengths of 14.2 mm (0.56 in.) or less and, (c) attempt to improve seal performance so that they are capable of operating under simulated helicopter transmission conditions with leakage rates less than 2cc/hr and at wear rates that would provide over 1100 hours of life.

During the technology review phase of the program, nine seal concepts were proposed for consideration. These concepts were:

- Universal Face Seal
- Floating Ring Seal
- Face Seal with Lubricant at Outside Diameter of Primary Seal
- Face Seal with Lubricant at Inside Diameter of Primary Seal
- Circumferential Seal
- Dual Element Split Ring Seal
- Floating Lip Seal
- Dual J Seal
- Hydrodynamic Lip Seal

From these concepts the circumferential seal and the dual element split ring seal were chosen for detail design, fabrication and test. The seals were designed for a shaft diameter of 13.6 cm. (5.355 inch) and were operated at 6600 rpm with a bulk lubricant temperature of 384 K (230°F). Radial runout of the input shaft was .0508 mm. (.002 inch) total indicator reading and seal lubrication for most cases was by splash and partial flooding.

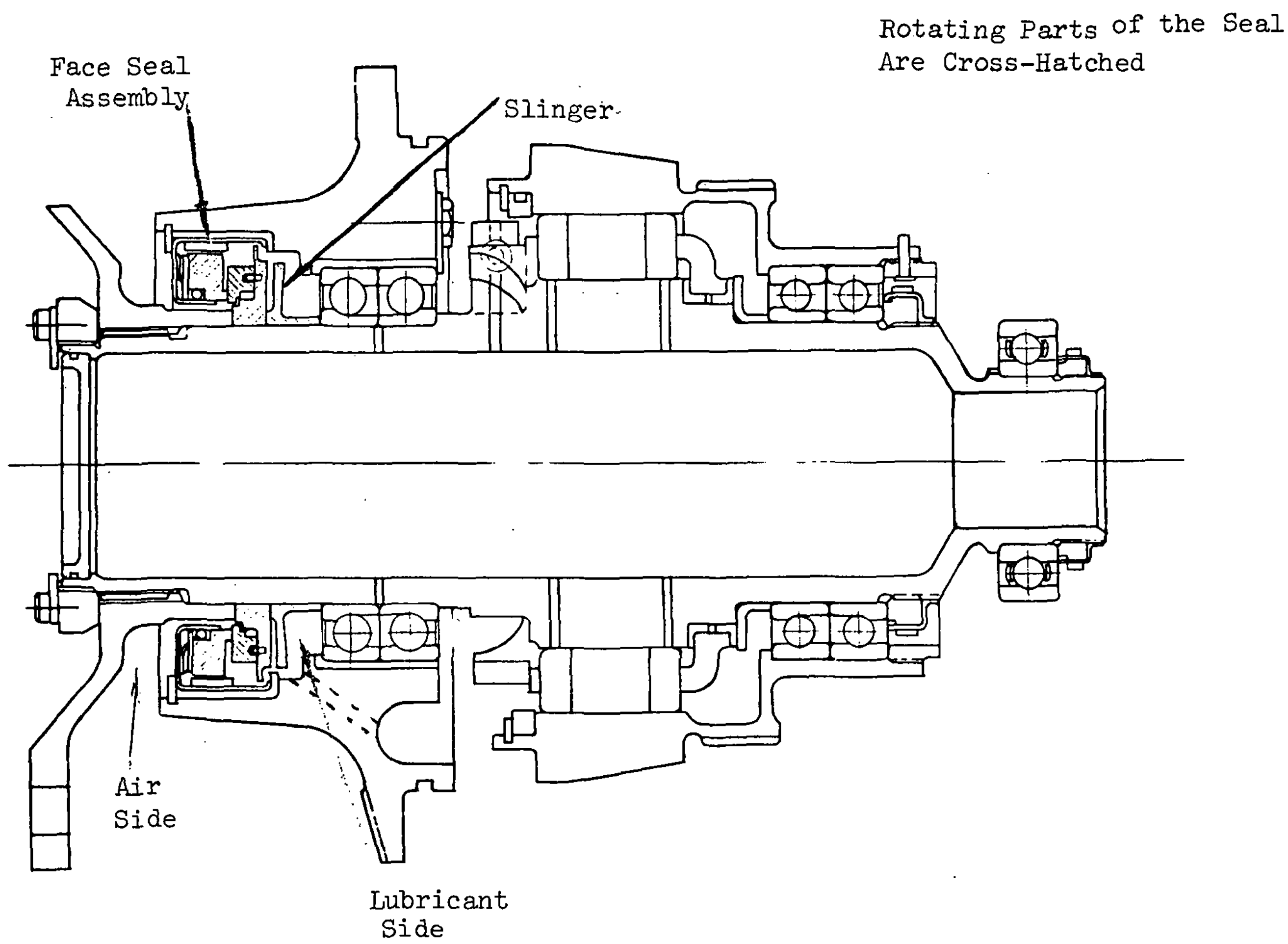

Figure 1. High Speed Face Seal Application.

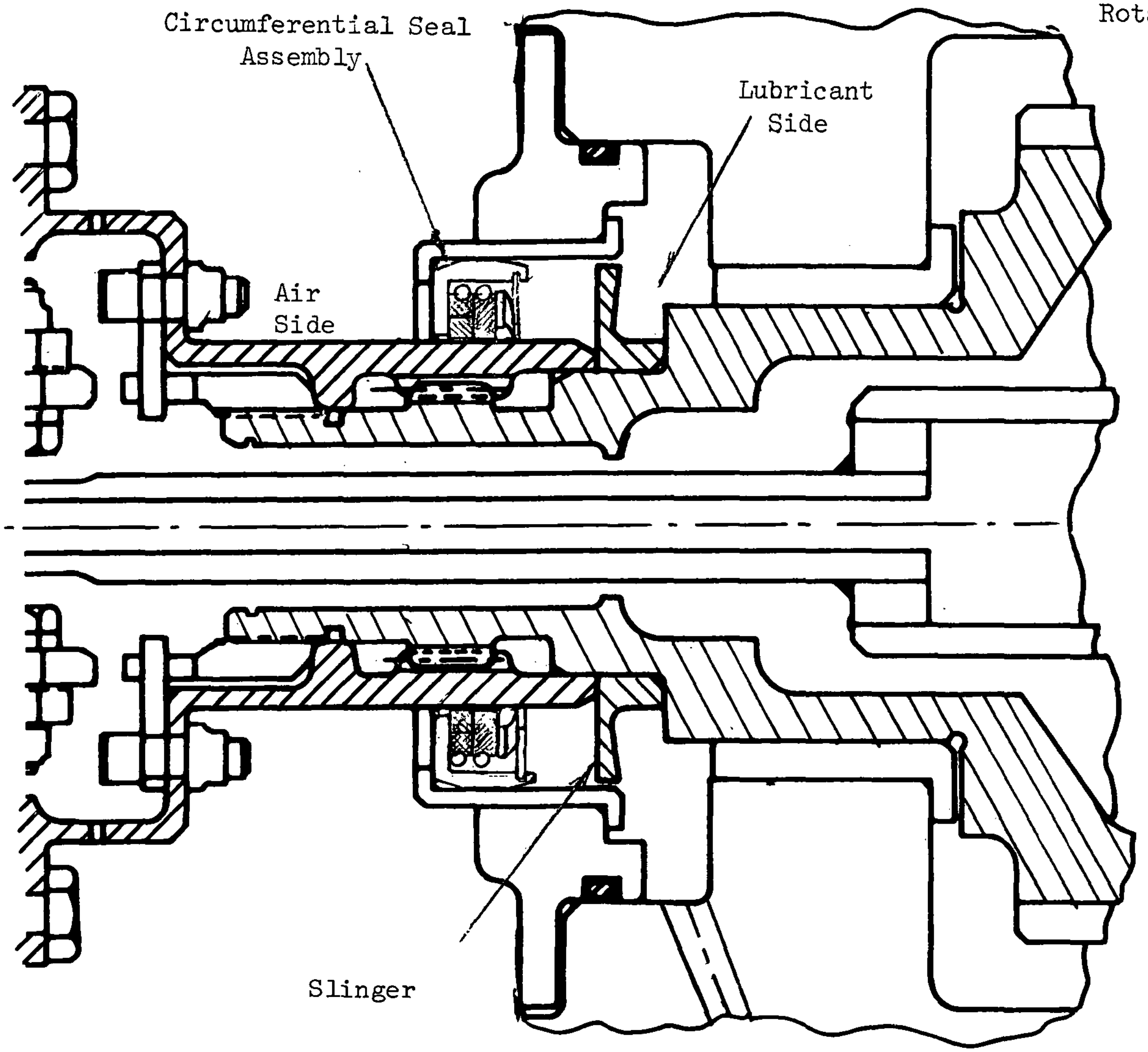

Figure 2. High Speed Circumferential Seal Application.

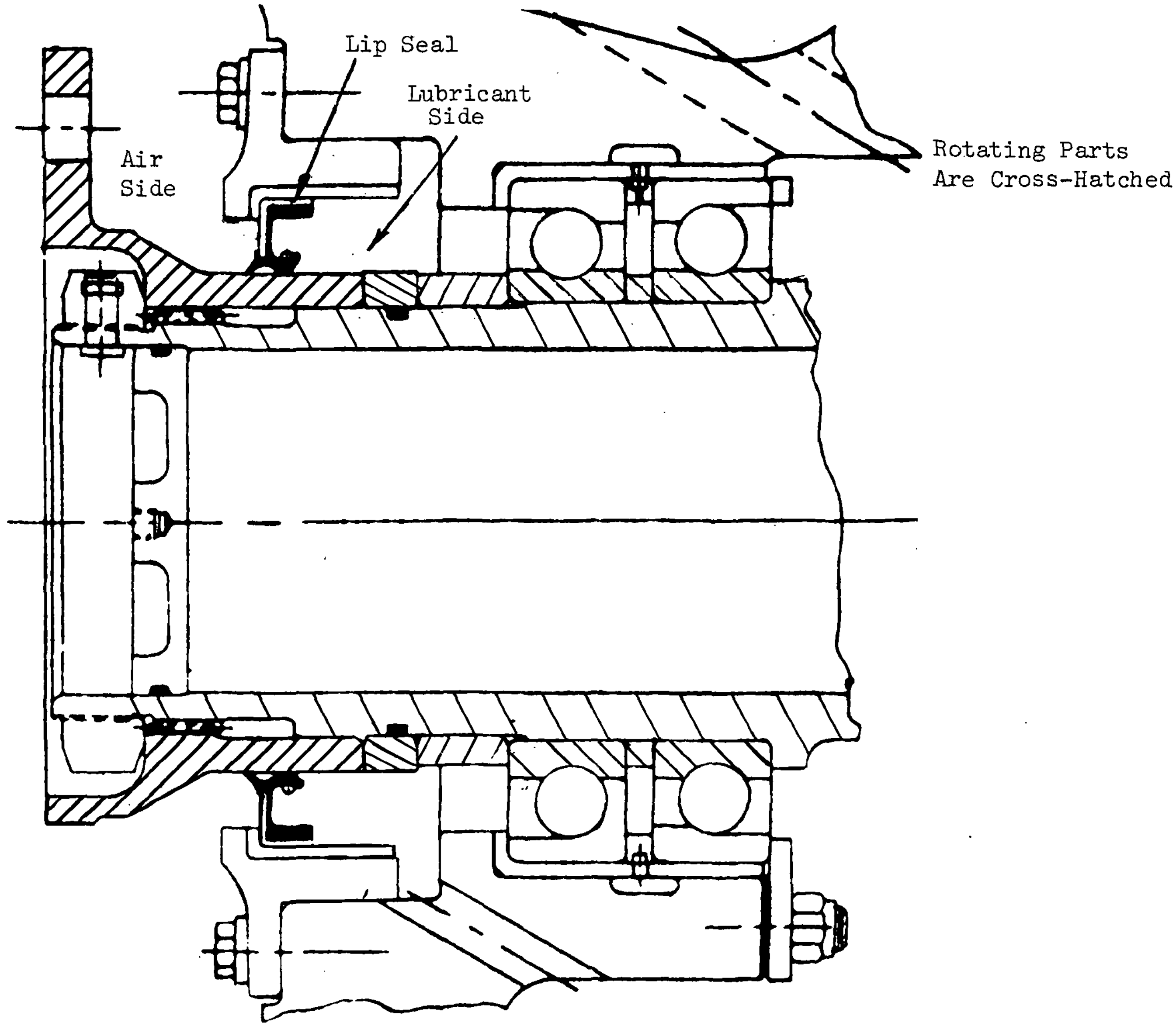

Figure 3. High Speed Lip Seal Application.

SECTION II

Design

A. Seal Concepts

The operating conditions given by the contract specifications were:

- . 6600 rpm shaft speed
- . 383 K (230°F) lubricant temperature
- . MIL-L-23699 lubricant

The test vehicle input assembly configuration were specified in the contract requirement and is shown in Figure 4. From the configuration the radial eccentricity and axial displacement at the seal were established by a stack-up of support component tolerances. The radial eccentricity is comprised of offset (static eccentricity) and radial runout (dynamic eccentricity), and is defined as the variation of true center of the housing to the true center of the shaft. The axial displacement is the variation in axial distance from housing to shaft locating surfaces.

Table I and Figure 5 show the geometric envelope, operating conditions and performance criteria of the input shaft test assembly.

From the considerations of these conditions and requirements, nine seal concepts potentially capable of successful operation were proposed to the NASA project manager. These concepts are shown in Figures 6 thru 14 with a brief discussion of the seal elements, their capabilities and limitations. From these nine concepts, two were selected for test; these two were the dual element split ring and the circumferential seal.

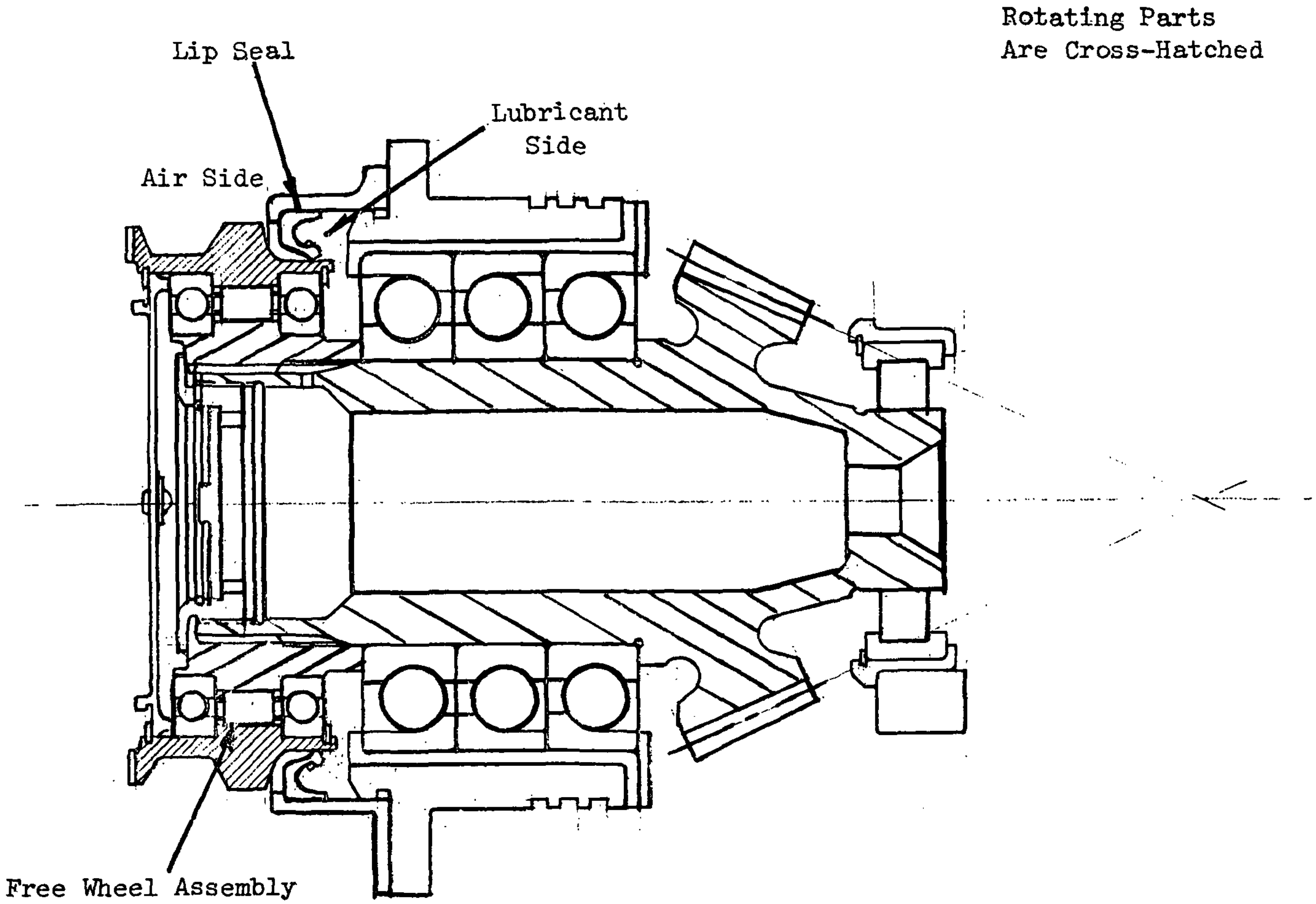

Figure 4 - Test Vehicle - Input Quill Assembly

TABLE I - APPLICATION PARAMETERS

Application:	INPUT SHAFT SEAL
Shaft Outside Diameter	
Shaft Diameter m.m. (inches)	136 (5.356/5.354)
Surface Finish Microns (Microinches) AA	.203 (8)
Lead (RH, LH, None)	Not Specified
Material	AMS 6260
Hardness	Chrome Plated
Radial Eccentricity (Static) m.m. (inch)	.406(.016)
Radial Runout m.m. (inch)	.17(.0067)
Out-of Roundness m.m. (inch)	Not Specified
Axial Displacement m.m. (inch)	± 1.07 (± .042)
Axial Runout m.m. (inch)	.0076(.0003)
Shaft Speed (Normal Operating)	6600 RPM
Direction of Rotation (viewed from air side)	Clockwise
Housing Bore	
Bore Diameter m.m. (inch)	165(6.5015/6.4985)
Depth m.m. (inch)	19 (.768/.713)
Surface Finish Microns (Microinches) AA	1.6 (63) MAX.
Material	MAG. ALLOY-AZ91C
Seal Performance	
Leakage c.c./hr.	1
Life	1100 Hours
Operating Fluid	
Type of fluid being sealed	MIL-L-23699
Temperature (Maximum Operating)	384 K (230°F)
Environment	DUST, MOISTURE

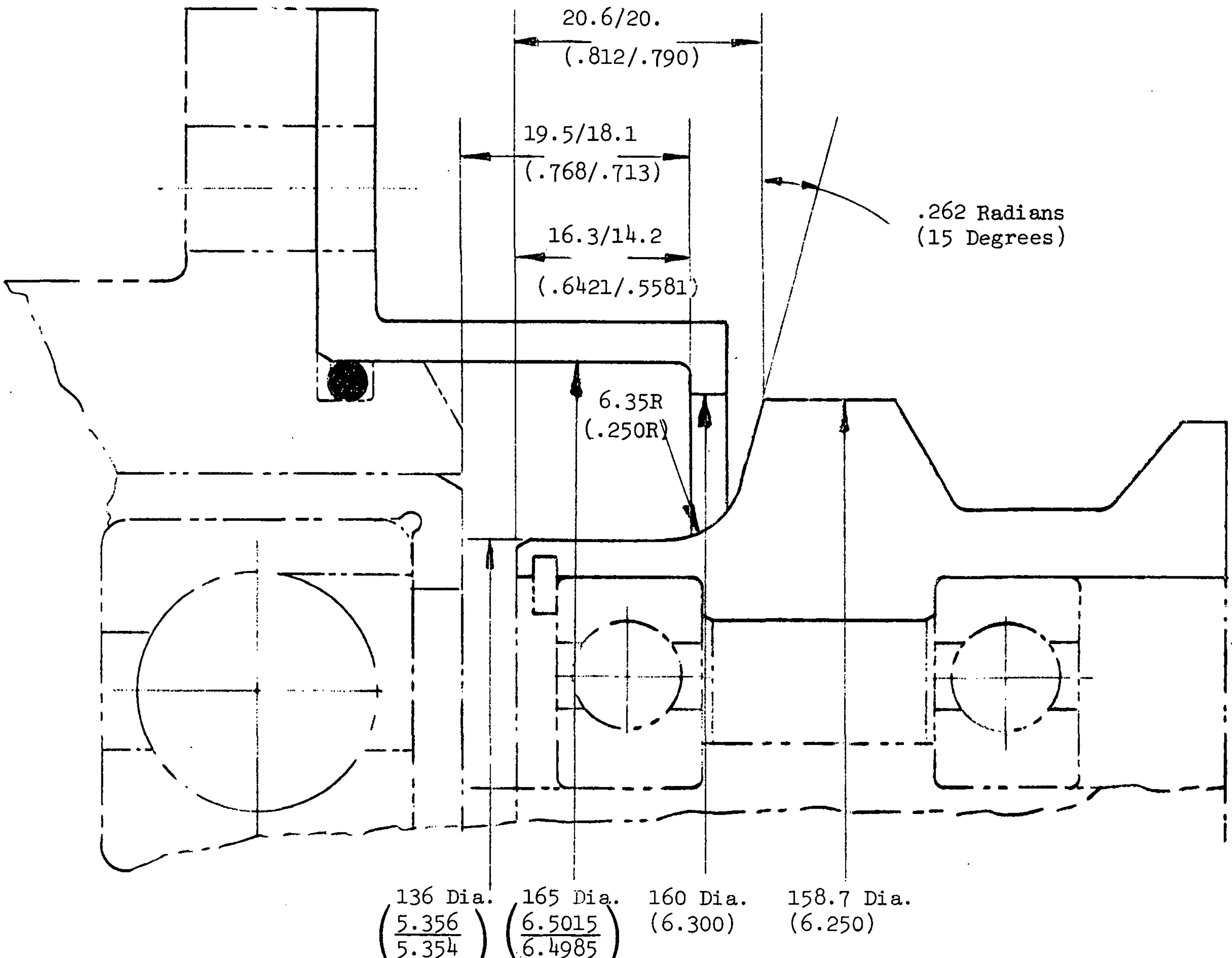

Figure 5. Seal Envelope Limitations. Dimensions are in m.m. (inch).

Dual Element Face Seal (Figure 6)

This seal is a self-contained (unitized) face type seal. The rotating elements consisting of a primary seal ring, usually carbon graphite, and a split ring, usually carbon graphite or another good bearing material. These rings are forced against hardened steel stationary flanges by a wave spring exerting approximately 6.88 N/cm^2 (10 psi) face pressure. The rotating components are driven by the radial clamping load of the split ring on the shaft.

Oil seeps through the split ring element to provide lubrication for the primary seal ring and hydrodynamic features such as recess pads which are sometimes used in the face of the split ring in order to provide improved lubrication. The secondary seal is an elastomeric "O" ring on the shaft. Oil holes in the oil side flange vent the seal cavity to prevent a build-up of lubricant.

One advantage of this seal is that the primary face loading is not affected by axial location which for the test assembly application is ±1.06 m.m. (±.042 inch).

This type seal has been used successfully in the transmission systems of Sikorsky Aircraft model S-55, S-56, S-58, S-61, S-64 and S-65 helicopters. Its two most recent uses are in the S-61 and S-65 auxiliary power plant clutches. Basic operating conditions for the S-65 clutch are:

- 8200 rpm
- 48.6 m.m. (1.875 in) shaft diameter
- 1.38 N/cm^2 (2 psi) oil pressure
- 373 K (210°F) maximum oil temperature

Design layouts revealed that use of this seal type in the test vehicle configuration of Figure 4 would require component changes or rework in order to provide sufficient room for the seal.

Floating Ring Seal (Figure 7)

This seal is also a face type seal. Dynamic sealing is accomplished between the air side housing flange and the carbon graphite nose piece (primary ring). The carbon graphite ring must be loaded against the hardened mating surface (flange) by pressure. If the pressure is not sufficient to seat the primary element the seal operates as a labyrinth with a gap of approximately .127 m.m. (.005 inch). Static sealing and primary element drive is provided by an elastomeric "O" ring (secondary seal) on the shaft.

The carbon graphite primary ring is a shrunk fit into a steel sleeve to control the thermal and centrifugal growth.

From data supplied by seal manufacturers, it was shown that this type seal

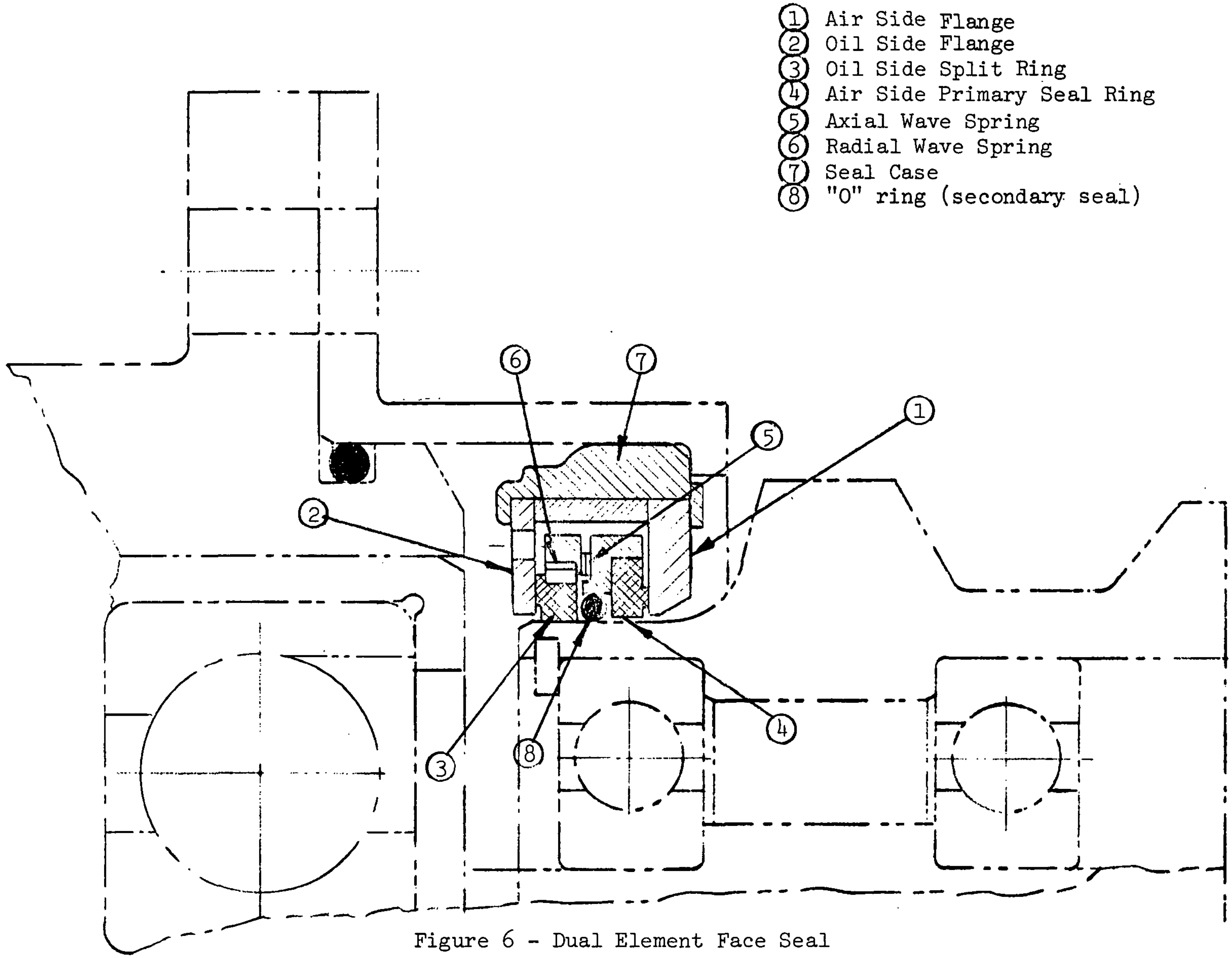

Figure 6 - Dual Element Face Seal

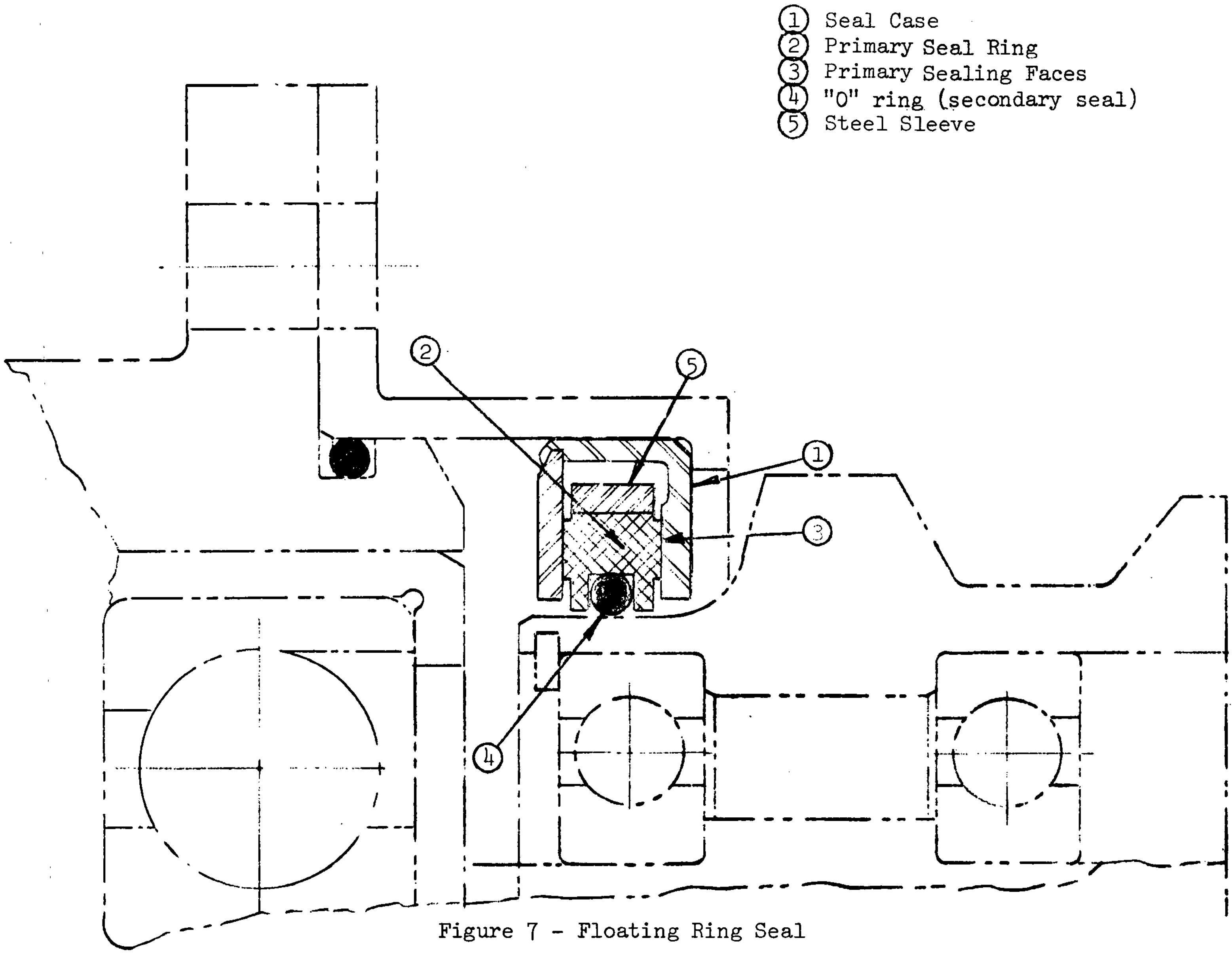

Figure 7 - Floating Ring Seal

is being used in a centrifuge application at the following conditions:

- 6200 rpm
- 140 m.m. (5-½ inch) shaft diameter
- seals oil at 34.4 N/cm^2 (50 psi) from water and sand at atmospheric pressure (leakage rate unknown)
- 356 K (180°F) oil temperature

This seal could be used in the input shaft test assembly without any rework to assembly hardware.

Face Seal (with lubricant at outside diameter of primary seal)
(Figure 8)

Standard face seals are the most common high speed seal used in industry. The seal consists of a non-rotating primary ring, usually carbon-graphite, which is flexibly attached to the case. The primary ring is allowed to move axially in the case, to adjust for dimensional variations, runout and thermal growth. The primary ring is loaded by a spring against a rotating mating ring. The mating ring material is usually a hardened steel or stainless steel. In this application, to avoid distortion to the mating ring and to allow for some shaft to housing misalignment, the mating ring assembly consists of a seal ring and clamping ring.

The use of this seal would require no rework or redesign of the input assembly.

This seal is used in most high speed seal applications in helicopter transmissions. Typical operating conditions for a successful face seal application are:

- 6023 rpm
- 90 m.m. (3.54 inch) seal nose diameter
- 1.38 N/cm^2 (2 psi) oil pressure
- 378 K (220°F) oil temperature

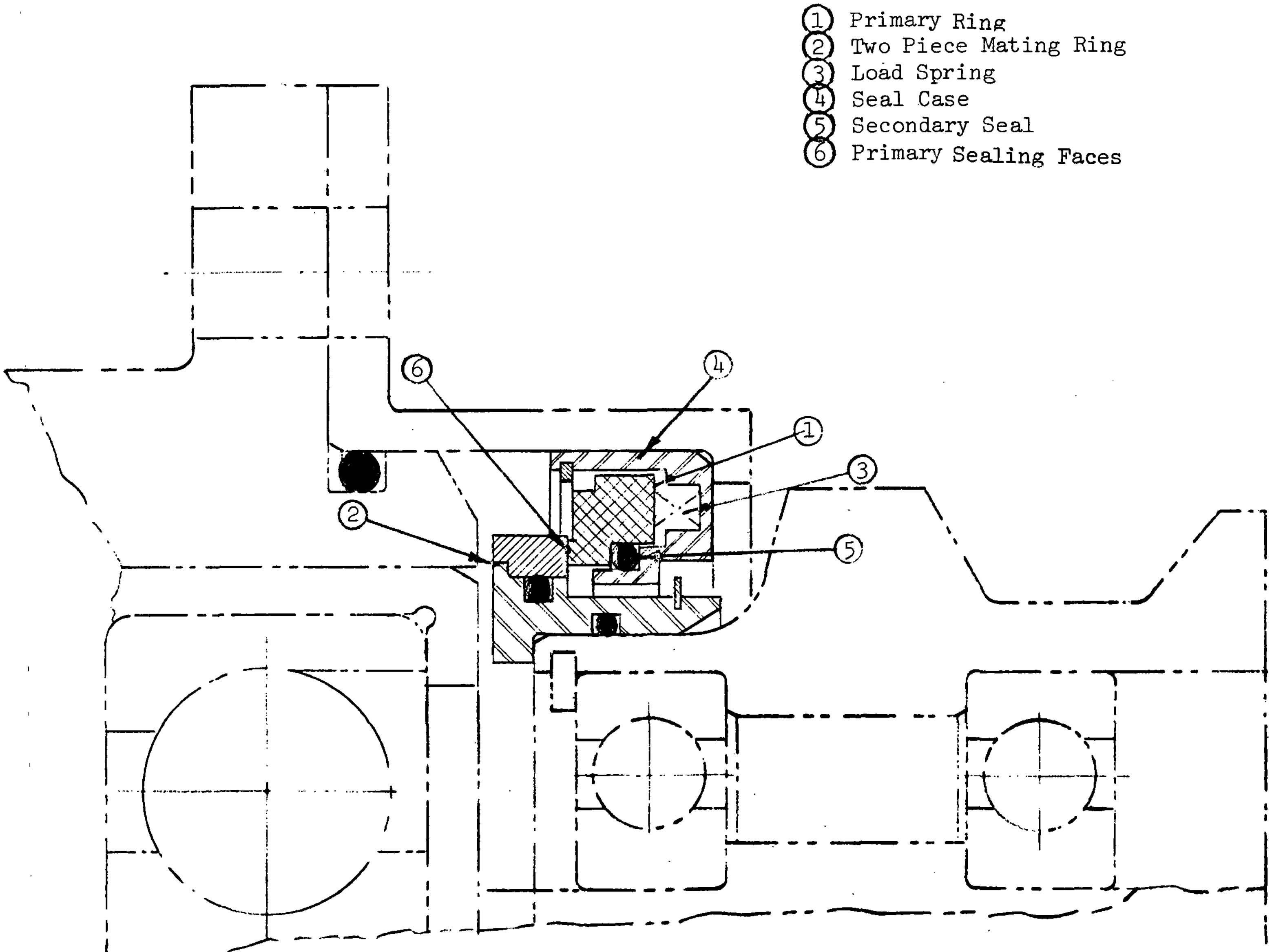

Figure 8 -- Face Seal with Lubricant at O.D. of Primary Seal

Face Seal (with lubricant at inside diameter of primary seal) (Figure 9)

This type seal has the same elements as the previously discussed face seal, except that the flexibly attached carbon-graphite primary ring is rotating and the mating ring is stationary. This construction is seldom used in transmission seals because centrifugal forces will increase leakage, while these forces act to prevent leakage in the previously discussed face seals. It is considered for this program because of the ease of assembly.

Circumferential Seal (Figure 10)

This type is a shaft riding seal with the dynamic seal occurring on a cylindrical surface. The primary seal rings are segmented carbon graphite rings, which are loaded radially to the shaft runner by garter springs and axially to the case flange by a wave spring.

The three segmented rings in this design are the primary ring, cover ring and back ring. The segments have straight cut gaps. (Other designs have one and two rings with different type gap construction.) Pins at the gap location provide anti-rotation locks.

Static sealing is accomplished by orientating the ring gaps to cover all leakage paths and by lapping all ring interfaces flat. Only slight relative motion occurs at the flange ring interface. This motion allows for static eccentricity and runout and requires the springs loads to be of proper relative magnitude to prevent frictional hang-up.

This type seal has approximately the same envelope requirements of a lip seal. It is independent of axial displacement of the shaft. No change in adjacent hardware would be necessary for its use.

Circumferential seals are currently used in many high speed, high temperature gas applications. It is currently being used for oil containment on the input shaft to the Sikorsky Aircraft S-61 main transmission at the following conditions:

- 20,000 rpm
- 42.8 m.m. (1.688 inch) shaft diameter
- 1.38 N/cm^2 (2 psi) oil pressure
- 389 K (240°F) oil temperature

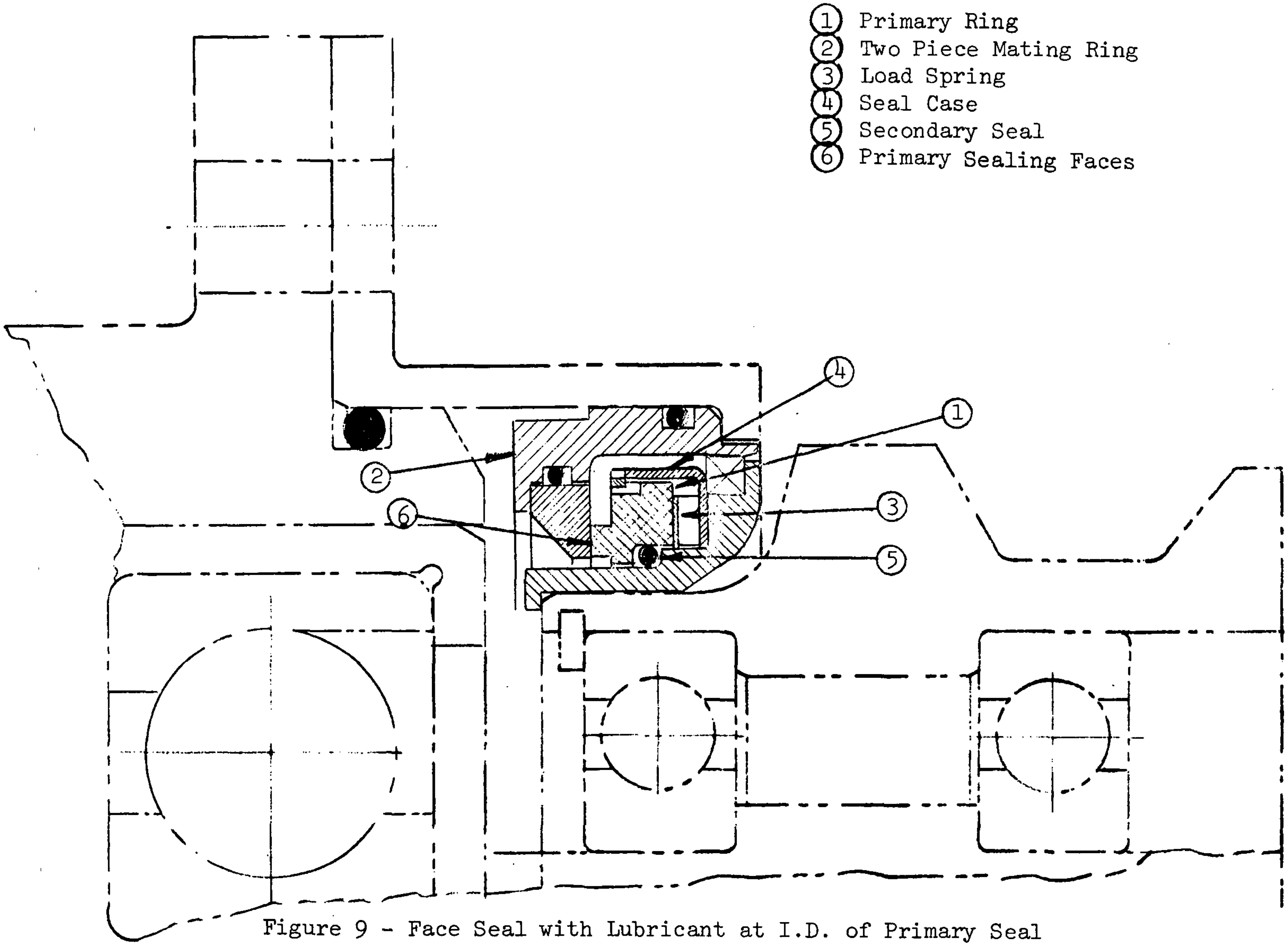

Figure 9 - Face Seal with Lubricant at I.D. of Primary Seal

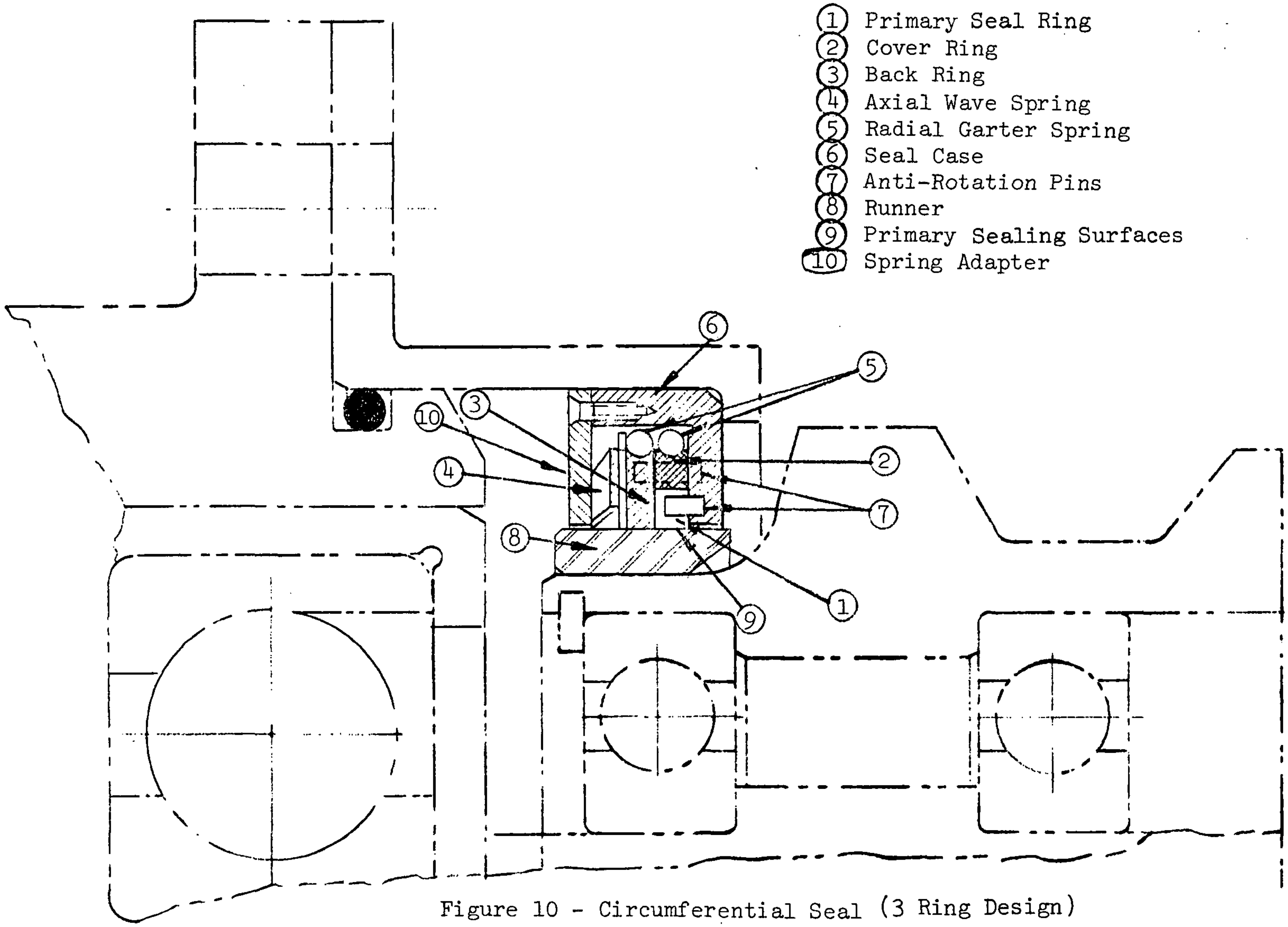

Figure 10 - Circumferential Seal (3 Ring Design)

Dual Element Split Ring Seal (Figure 11)

This type seal consists of stationary sleeve fixed to the housing, a rotating carrier attached to the shaft and two expanding split rings. The outer ring is teflon with a step cut joint while the inner expander ring is stainless steel with a straight cut gap. The radial load from the expander ring must be sufficient to seat the outside diameter of the teflon outer ring and prevent it from rotating.

In the low pressure environment of a helicopter transmission it is not expected that hydraulic axial loading will be high enough to seat the ring on the carrier groove side to form a positive contact dynamic seal. Therefore the mode of sealing is a close clearance labyrinth established by the ring and carrier groove sides.

It is necessary that the sleeve material be a hardened steel or stainless steel since some axial creep of the ring will cause wear on the sleeve bore. The carrier should be hardened since some occasional contact between the ring and carrier groove will occur. The sleeve has oil holes between the rings to permit leakage past the first ring to be drained back into the gearbox.

This seal design can be installed into the present envelope without any rework to other gearbox components.

A three ring seal design has been successfully tested in a high speed application in a helicopter production transmission for 500 hours at the following conditions.

- 13600 rpm
- .689 N/cm^2 (1 psi)
- 114 m.m. (4.5 inch) sleeve bore diameter
- 383 K (230°F) oil temperature

Floating Lip Seal (Figure 12)

This design consists of molded teflon lip which is pressed against the shaft by a secondary teflon element loaded by a garter spring and seated internally against the case sides by a wave spring.

The internal seal elements, including the primary lip and secondary lip, float radially in the seal case. This feature allows the seal to compensate for any shaft to housing offset (static eccentricity).

The seal can be made to conform with the dimensions of the present elastomeric lip seal used in the application. There has been no prior experience of operating this seal at conditions similar to the application under investigation.

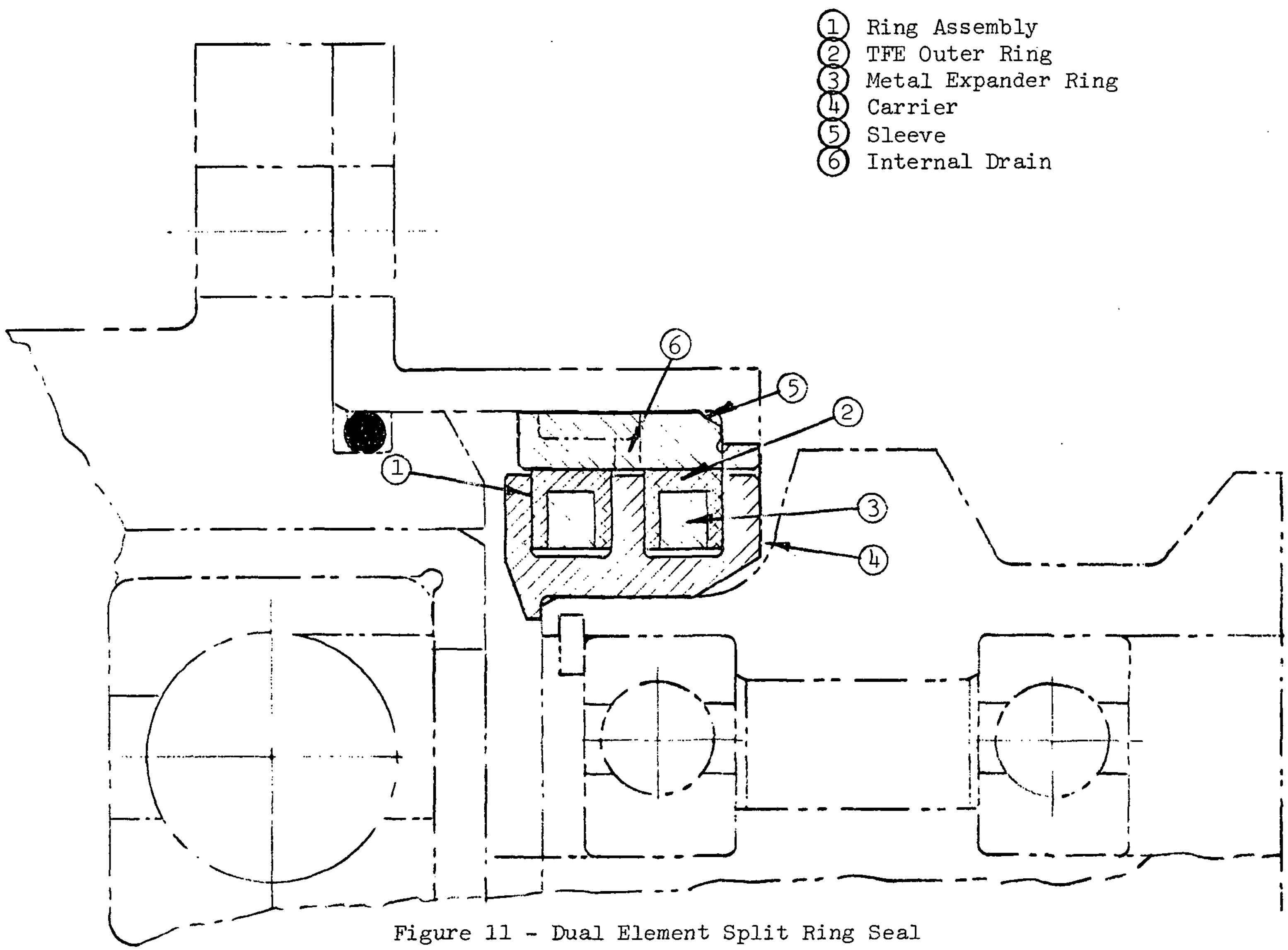

Figure 11 - Dual Element Split Ring Seal

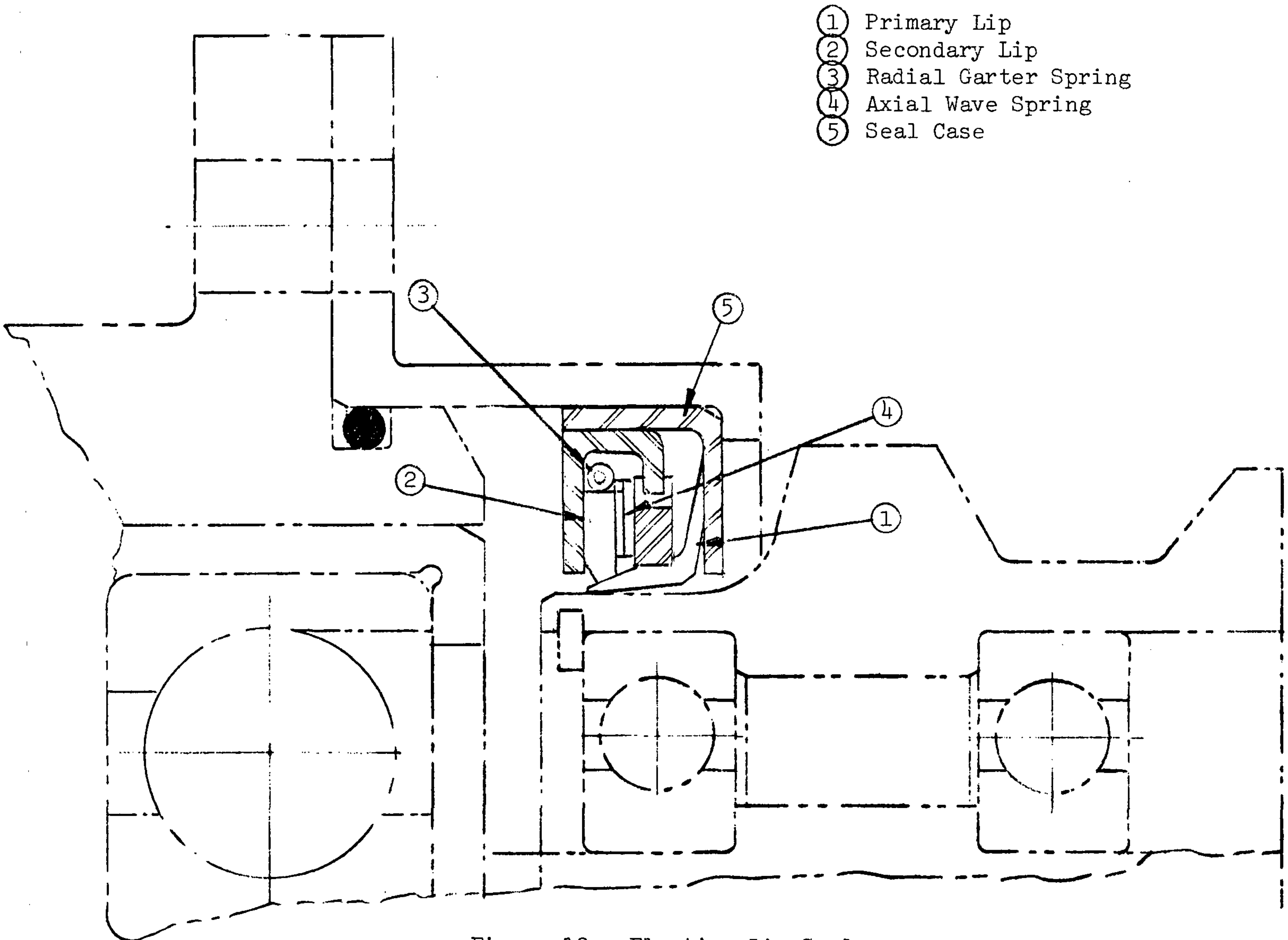

Figure 12 - Floating Lip Seal

Dual J Seal (Figure 13)

This design is similar to the floating lip seal. The primary seal element is a molded teflon lip which is pressed against the shaft by a secondary teflon lip and a garter spring. Static sealing is accomplished by an "O" ring interference seating both teflon element against the seal case sides. As with the floating lip seal the internal elements are not restrained in the radial direction.

The envelope for this type seal is smaller than a standard lip seal. Its basic advantage over the floating lip seal is its simplicity and low cost. No previous experience at similar operating conditions has been reported.

Hydrodynamic Lip Seal (Figure 14)

This type seal is similar to a conventional lip seal except for the addition of ribs which are molded on the air side surface. These ribs which are helix flutes approximately .0508 m.m. to .1016 m.m. (.002 to .004 inch) high at an angle of approximately .523 radians (30 degrees), establish a full fluid film at the lip contact point and pump back any oil seeping past the primary lip. The lip is completely molded which eliminates some of the manufacturing inaccuracies incurred during the trimming operation performed while forming the lip in older designs.

The lip material which is molded to a stainless steel case is a flouroelastomer which has a higher temperature capability (477 K (400°F)) than the present silicone lip seal material.

The seal design is independent of axial location but its speed potential is limited by radial eccentricity which will cause uneven radial lip loads and wear.

The seal will be made to the same envelope specifications of the present seal used in the application.

This type seal is currently being used extensively in the automotive industry for oil containment and in other applications where high reliability sealing is necessary. Its speed potential is greater than a conventional lip seal, although its speed limit is unknown and dependent on factors such as pressure and eccentricity. A successful high speed helicopter transmission application currently operating with a hydrodynamic lip seal has the following operating conditions:

- 6600 rpm
- 91.5 m.m. (3.600 inch) shaft diameter
- 1.38 N/cm^2 (2 psi) oil pressure
- 378 K (220°F) oil temperature

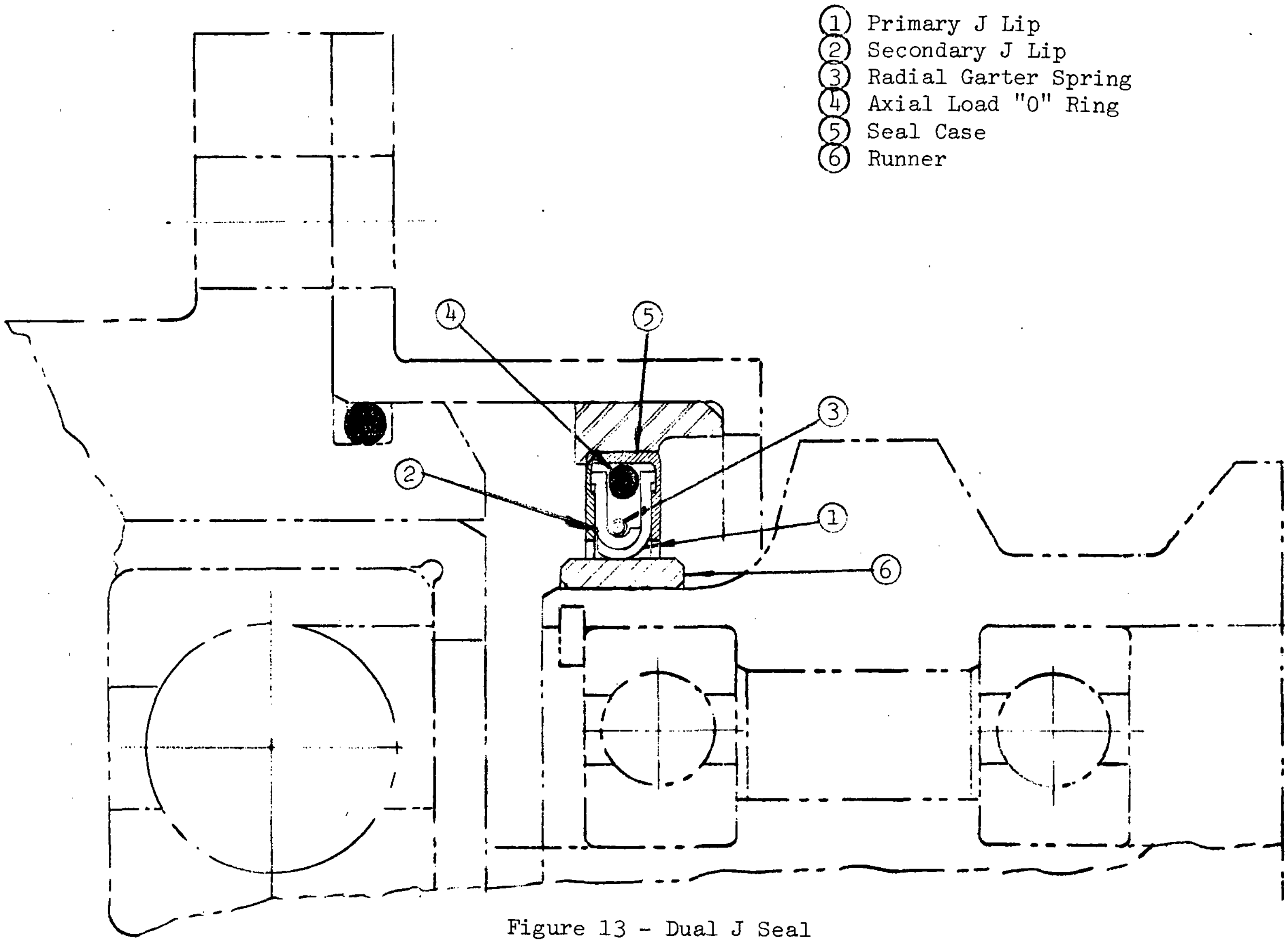

Figure 13 - Dual J Seal

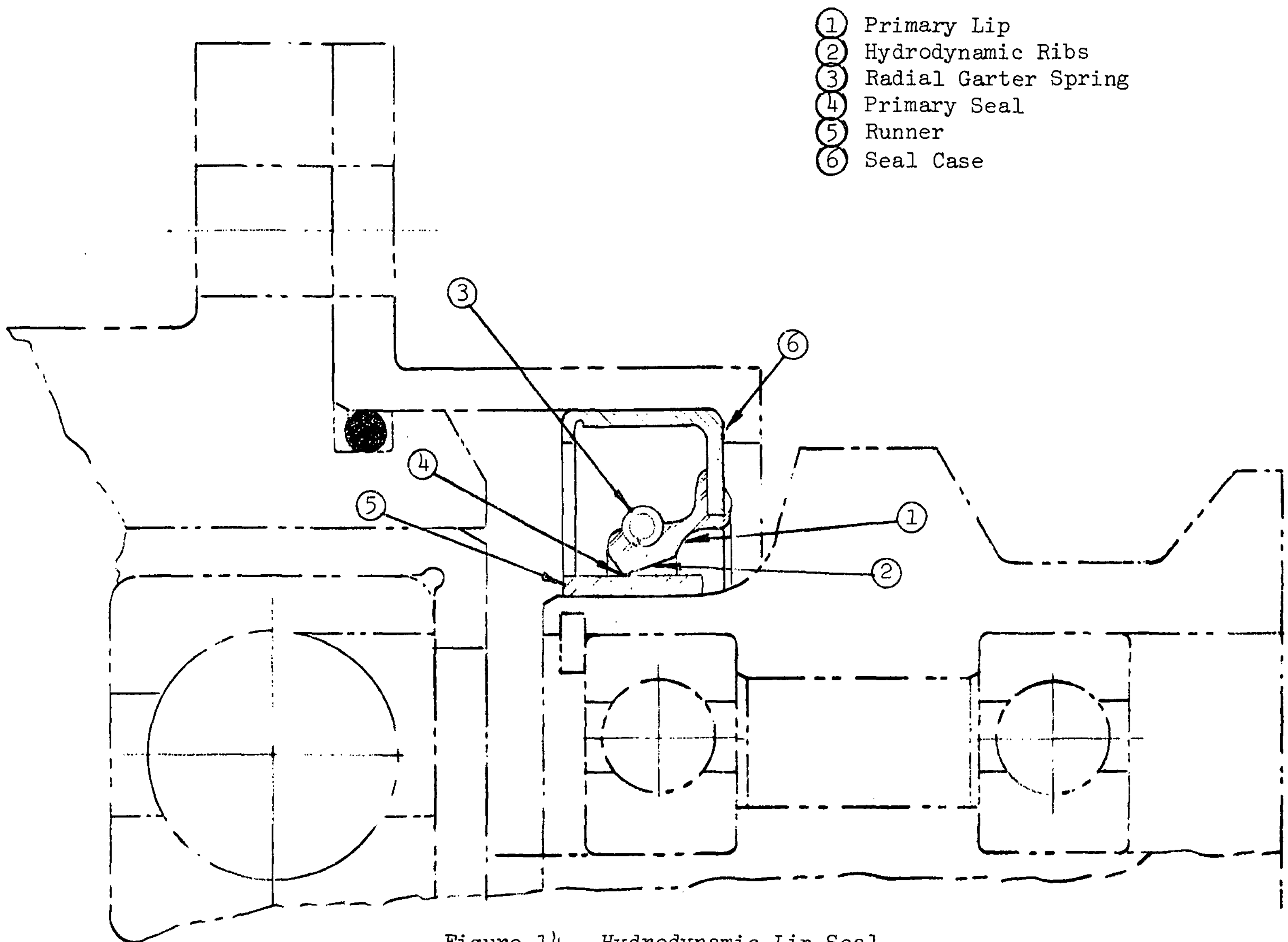

Figure 14 - Hydrodynamic Lip Seal

TABLE 2 - SUMMARY OF PROPOSED SEAL DESIGNS

Seal	Advantage	Disadvantage	Reason for Selection or Rejection
Dual Face	Carbon-graphite primary ring has good wear and high temperature properties. Face loading independent of axial tolerance stack-up. Independent of radial eccentricity.	Dependent on split ring for seal drive. Unknown effect of rotational inertia. Relatively expensive.	Rejected because axial length exceeded 14.2 m.m. (.56 inch).
Floating Ring	Carbon graphite material has good wear characteristics and thermal properties. Relatively inexpensive.	Will not be a positive contact seal in low pressure environment. Drive of primary element accomplished by "O" ring squeeze.	Rejected because of leak potential under flooded conditions.
Face Seal (with lubricant at O.D. of primary seal)	Positive contact seal with much experience in similar applications. Primary seal has high temperature properties.	Face loading dependent on axial location. Relatively expensive.	Rejected because of axial length.
Face Seal (with lubricant at I.D. of primary seal)	Has the same characteristics of face seal discussed above.	Centrifugal force tends to drive lubricant thru the primary seal.	Rejected because of leakage tendency.
Circumferential	Carbon-graphite primary elements are positive loaded radially and axially. Seal has high temperature limit because of no elastomeric parts. Independent of axial location and offset.	Sealing sensitive to radial runout. More effective as a gas seal than a liquid seal. Relatively expensive.	Selected because of small axial length, prior satisfactory experience and ease of assembly.
Dual Element Split Ring	TFE shrouded ring has good wear and thermal characteristics. Effective labyrinth with drainback between rings.	Not a positive contact seal at low pressure. Relatively expensive.	Selected because of small axial length and ease of assembly.
Floating Lip	TFE element has good wear and high temperature characteristics. Positive contact primary element loaded radially and axially. Independent of axial location and offset.	TFE element not capable of following shaft motion or irregularities. Relatively expensive.	Rejected because of leakage potential at high speed and dirt sensitivity.
Dual J	Same properties as floating lip seal. Relatively inexpensive.	TFE primary element will not conform to shaft irregularities or motion.	Rejected because of leakage potential at high speed and dirt sensitivity.
Hydrodynamic Lip	Molded ribs will retard any seepage past the primary lip. Ribs will help establish a full fluid film. Flouroelastomer capable of high temperature. Relatively inexpensive.	Flouroelastomer not suited for low temperature operation. Reliability of lip seals at high surface speeds is questionable.	Rejected because of leakage potential after several hundred hours and dirt damage sensitivity.

B. Seal Detail Design

The two designs that were selected from the nine candidate seals were subjected to a detailed study to establish the design parameters of each seal design.

1. Dual Element Split Ring Seal

The detail study and design of the split ring seal was based on the experience obtained in testing performed on a three ring design at similar operating conditions in the H-53 model helicopter nose and main gearboxes. It was determined from this effort that:

1) Two piece (teflon shrouded) step-cut rings controlled leakage more efficiently than one piece ductile iron straight cut rings.

2) The sleeve and carrier should be hardened to permit occasional ring creep or rubbing.

3) The sleeve and carrier should have similar thermal coefficients of expansion to maintain a constant radial clearance.

4) The split rings will effectively seal a lubricant spray but will leak excessively if exposed to a flooded condition.

With these parameters established the detail design of the seal was undertaken. Due to space limitations, the number of rings was limited to two. A ring outside diameter of 15.9 cm (6.25 inch) was chosen to give sufficient thickness to the sleeve for heat treating purposes. Due to the small space envelope a narrow ring width was desirable. A minimum total ring width of 4.60/4.55 m.m. (.181/.179 inches) and expander ring width of 2.49/2.44 m.m. (.098/.096 inches) was considered to provide sufficient material thickness of the carrier groove sides for structural and manufacturing purposes. Data established through previous experience indicates that 1.75 newton per cm. (1 lb. per inch) of circumferential radial ring load is a good design allowable, a radial wall thickness of the ring was then calculated by the following equation:

$$U.L. = .1414\ E\ g\ b/D(D/d-1)^3$$

where
- U.L. = unit load, newtons
- E = modulus of elasticity, newtons/cm^2
- g = free gap less clearance, cm
- b = width, cm
- d = wall cm
- D = sleeve bore, cm

The ring assembly is shown in Figure 15.

The ring assembly consits of a TFE outer ring and stainless steel inner ring expander. The outer ring is seated against the sleeve bore by the radial force of the inner ring. The outer ring is step cut with the step mating surface machined flat to minimize internal leakage in the axial direction. The inner ring is a straight cut stainless steel (17-4PH) expander which in conjunction with the outer ring retards internal leakage in the radial direction. The gap of the inner ring and of the outer ring must be staggered in order to form an internal seal.

The design of the sleeve and carrier require a hardenable stainless steel to reduce wear and corrosion. A 440C stainless steel was selected with a hardness of Rockwell C56-60. The radial gap between the sleeve and carrier was specified to prevent interference at maximum eccentricity. The minimum radial gap between the sleeve and carrier must be greater than the anticipated radial eccentricity (offset plus runout). Although the runout was calculated to be .17 m.m. (.0067 inch), it was reported that .430 m.m. (.017 inch) runout could occur with the test assembly. The reported maximum runout, .432 m.m. (.017 inch) combined with a maximum static eccentricity, .406 m.m. (.016 inch), Table I, and clearance of .127 m.m. (.005 inch) produced a minimum radial clearance of .965 m.m. (.038 inch). Special design features of the two components are discussed below:

(1) Carrier

- To prevent dimensional and geometric change of the freewheel unit support bearing,(Figure 16), drive of the carrier was accomplished by an "O" ring packing with a radial squeeze of 25%.

- The carrier groove was .165/.241 m.m. (.0065/.0095 inch) wider than the ring to allow for differential thermal expansion between the carrier groove sides and the teflon outer ring sides.

- A .406 micron (16 microinch) AA surface finish of the groove sides was specified for minimizing wear during periods of rubbing contact.

- The groove depth was 1.6 m.m. (.063 inch) greater than the ring inside diameter to prevent contact of the ring with the bottom the groove at maximum eccentricity.

(2) Sleeve

- The sleeve bore surface finish was .406 microns (16 microinch) AA to permit ease of assembly and axial movement of the rings.

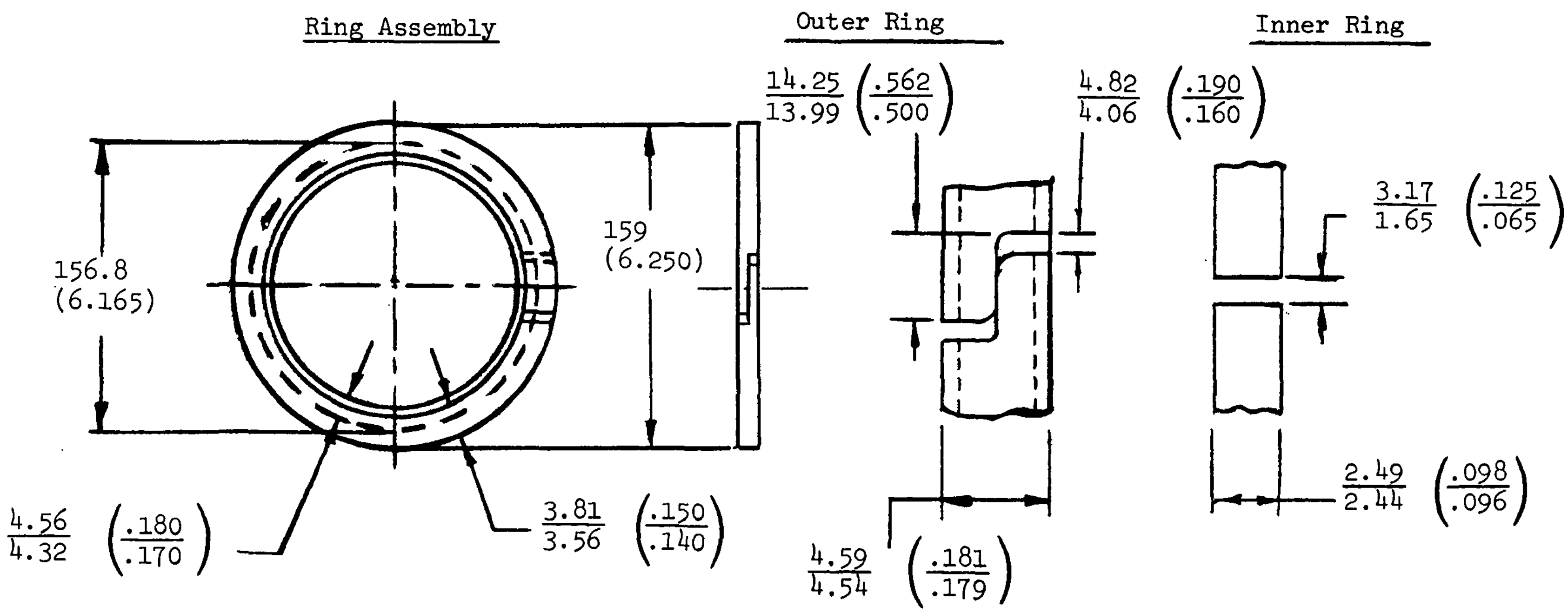

Figure 15. Two Piece Step Cut Ring. Dimensions are in m.m. (inch).

- An annulus at the sleeve bore was incorporated to disrupt the flow of leakage past the first ring.

- Radial holes were used to facilitate internal seal drainage, if required.

- The sleeve was held and sealed on the outside diameter by "O" ring packings.

- The width of the sleeve was specified to insure full contact of the ring outside diameters at the extremes of the axial operating range.

Figure 16 illustrates the assembled seal.

2. Circumferential Seal

The study and design of the circumferential seal involved integrating successful practice of circumferential gas seals and successful design features of the H-3 main gearbox input seal. A three ring design was selected over a single ring or double ring design because of the ease of manufacture and the vast experience with this design.

Although a hard runner of at least Rockwell C55 is desirable for a circumferential seal, the runner used with an existing lip seal was selected for economic reasons. It has the following characteristics:

- Stainless steel material.

- Plunge ground with a .254 to .508 micron (10 to 20 microinch) AA finish.

- 136 ± .0508 m.m. (5.349 ± .002 inch) inside diameter.

- 139.5 m.m. (5.481 inch) outside diameter.

- Hardness of Rockwell C30.

The material of the primary seal rings was carbon graphite. Each ring was cut into three 2.1 radian (120 degree) segments with an anti-rotation pin fixed to the case flange located at the segment gaps. A snap ring (wide radial-width) was used to retain the loading spring.

The carbon graphite material used in the seal has the following properties:

- Scleroscope hardness of 85
- Compressive strength of 22350 N/cm^2 (32500 psi)
- Transverse strength of 6890 N/cm^2 (10,000 psi)
- Tensile strength of 5160 N/cm^2 (7,500 psi)

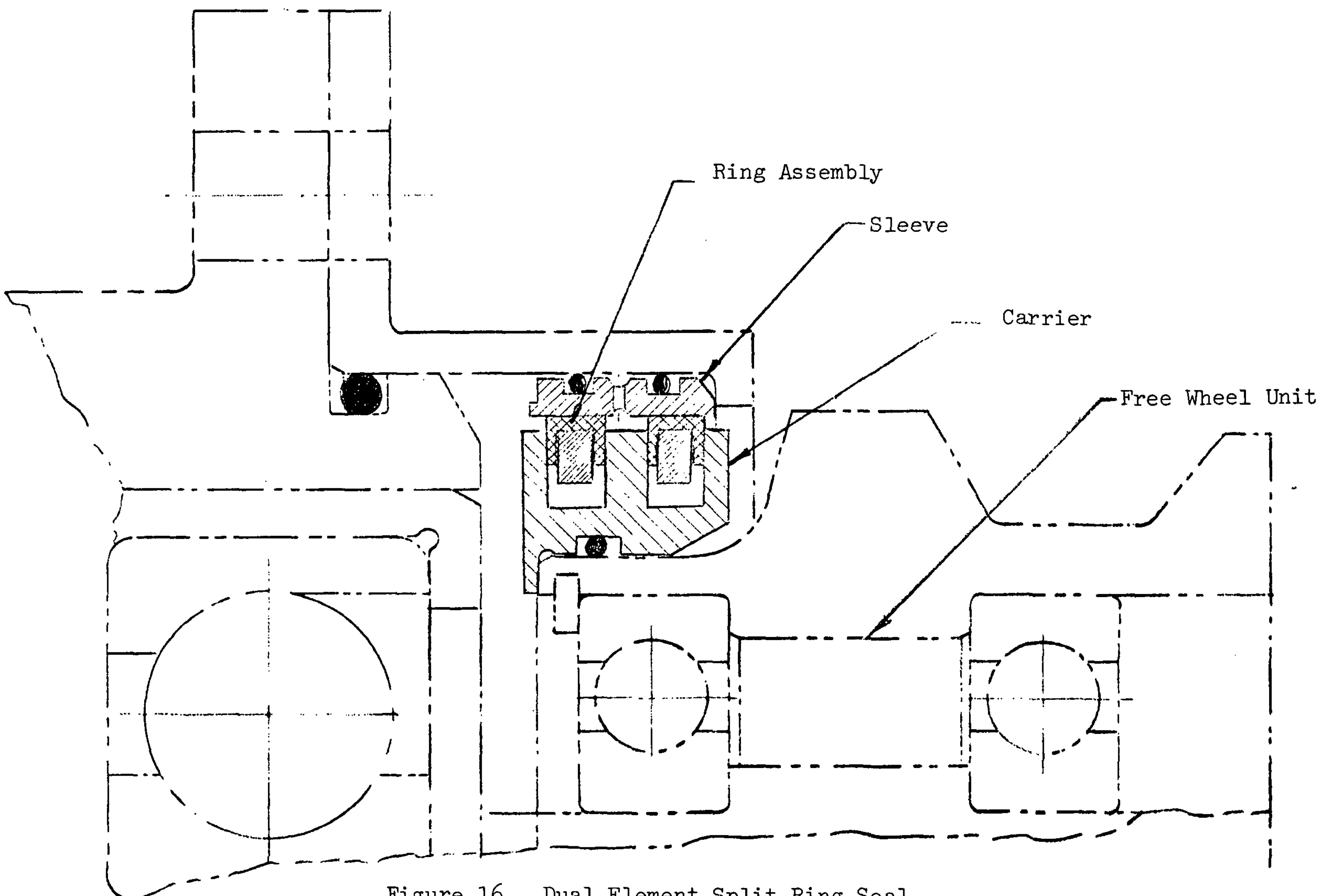

Figure 16. Dual Element Split Ring Seal.

- Modulus of elasticity of 2.13×10^6 N/cm^2 (3.1×10^6 psi)
- Oxidizing temperature limit of 477 K (400^{o}F)
- Coefficient of thermal expansion 1.665 m./m/ K (3.0×10^{-6} in/in/^{o}F)
- Permeability of 0.01×10^{-6} darcies

The seal case material was 6061-T6511 aluminum having a thermal coefficient of expansion of 7.0×10^{-6} m/m/K (12.6×10^{-6} in/in/^{o}F) and anodized per AMS-2470. Aluminum was used as a seal case material to reduce the housing-case inference fit by choosing transmission housing and seal case coefficients of thermal expansion as close as possible (the magnesium transmission housing has a thermal coefficient of expansion of 7.775×10^{-6} m/m/K(14.0×10^{-6} in/in/^{o}F)).

The sides of the carbon-graphite primary rings were lapped flat to .89 microns (35 microinches). The outside diameter of the primary ring and the inside diameter of the cover ring where machined round and tested to be light tight. This is necessary to minimize seal leakage.

The gap of the primary ring was machined at an angle, as shown in Figure 17, to hydrodynamically pump back leakage to the oil side of the seal.

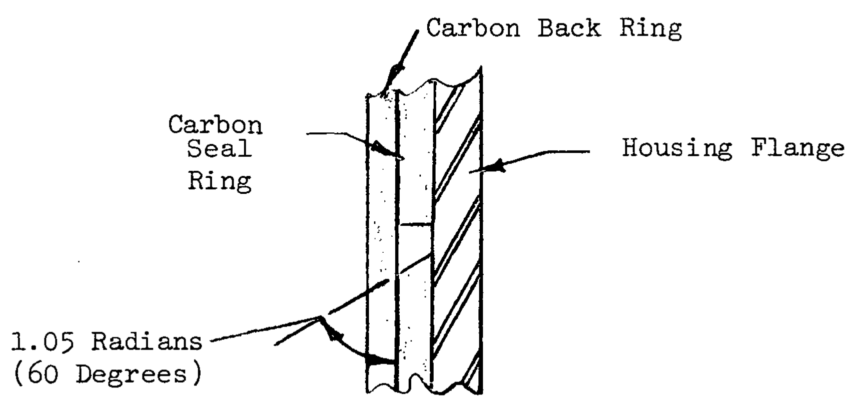

Figure 17. Gap Configuration of Primary Ring (View of Bore)

The primary carbon graphite rings were designed to a minimum width per ring of 2.41/2.29 m.m. (.095/.090 inch), to reduce the mass of the rings. With a reduced mass, the rings can respond quickly to shaft radial runout.

The garter springs were specified to have a radial load of .91 newtons per cm (.0521 lbs. per inch) of circumference. The installed wave washer axial load was 26.6 newtons (6 lbs.). The circumferential seal assembly is shown in Figure 18.

C. Test Rig Design

The test rig design, built under the subject contract, incorporated both aircraft and commercial components. The aircraft components included an engine drive shaft and an input quill assembly consisting of a bevel gear pinion shaft, support bearings, bearing and seal housings and a freewheel unit. The aircraft components were enclosed in the test facility housings (quill housing and drive shaft housing) and driven directly by a 11200 watt (15 HP) variable speed motor as shown in Figure 19. The speed of the test input assembly could be varied from 2400 rpm to 12000 rpm. The test rig incorporated the following features:

1) A lubrication system, as shown in Figure 20, which thermostatically controlled oil temperature and allowed for two independently controlled lubricant inputs (to the bearing housing and to the freewheel unit).

2) A motor reverse switch which was incorporated to simulate the freewheel mode of the input assembly.

3) A contaminent spray tube to introduce abrasive dirt to the seal area.

4) Lubricant baffles in the drive shaft housing to collect seal leakage.

The test rig was instrumented with copper constantan thermocouples to determine temperatures of the seal area and bearing housings. Two oil flow rate pickups (for the bearing housing lubricant and the freewheel unit) and pressure pickups (in the seal area) were installed to determine the effect of lubricant flow rate on seal pressure and leakage.

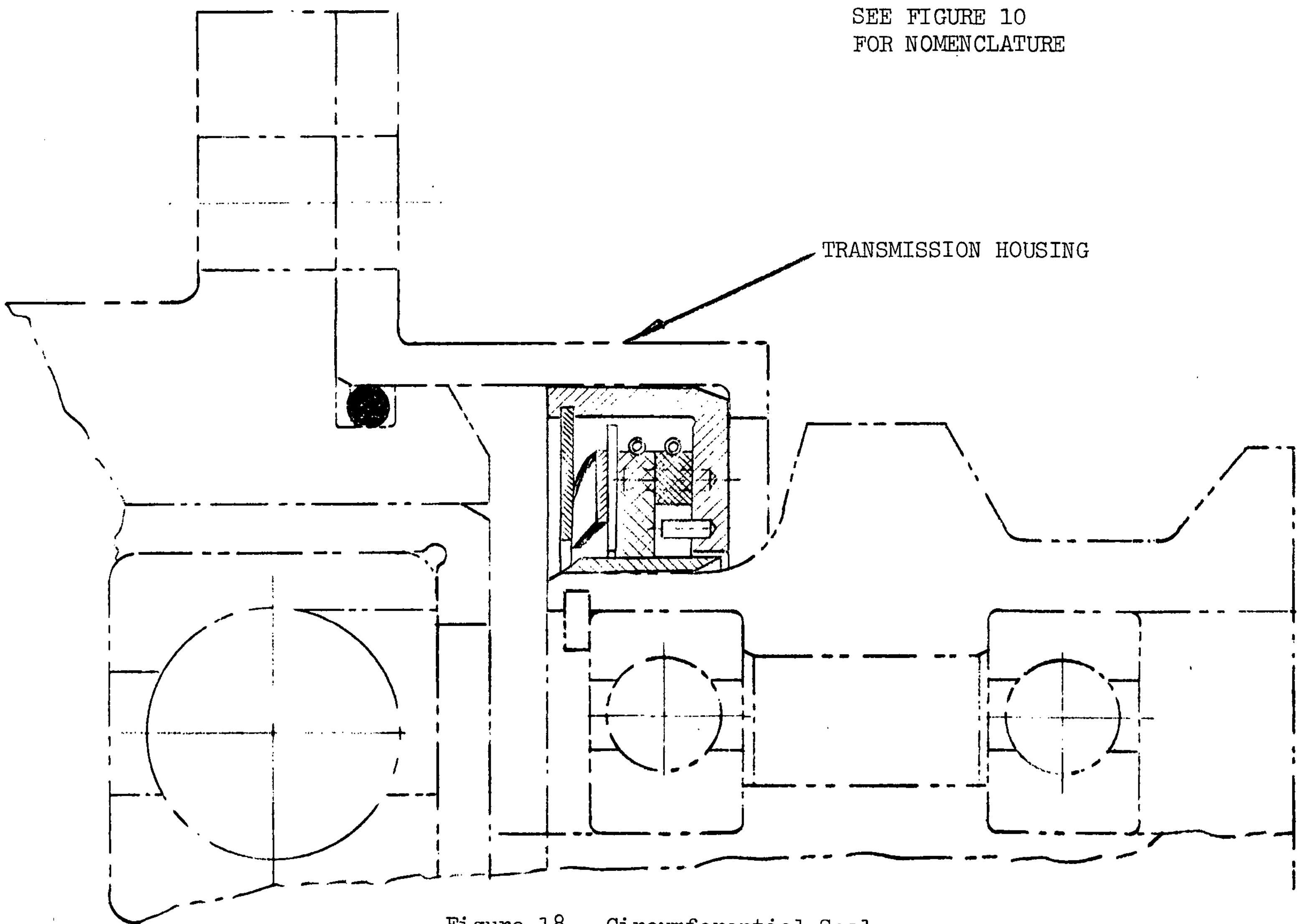

Figure 18. Circumferential Seal

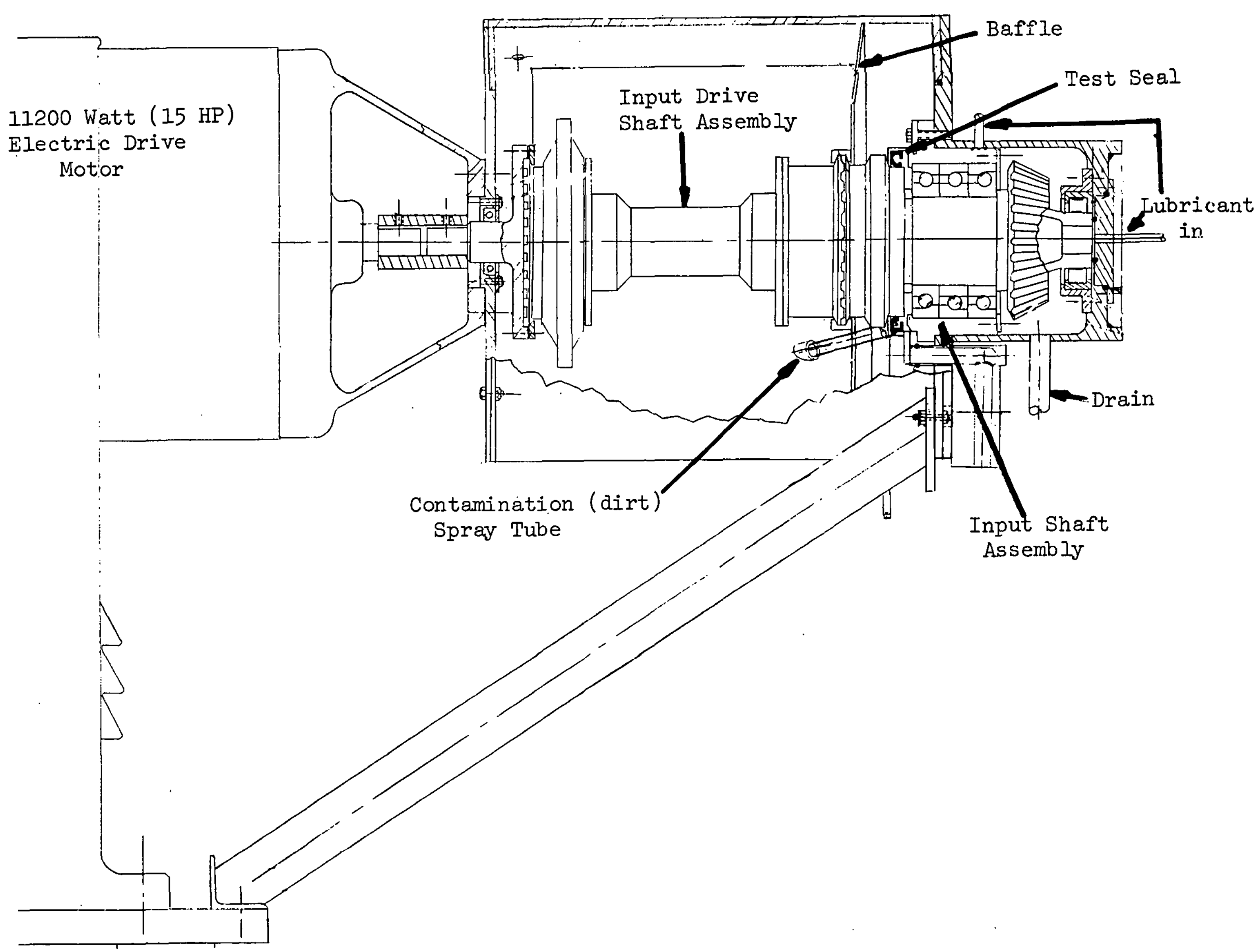

Figure 19. Seal Test Rig.

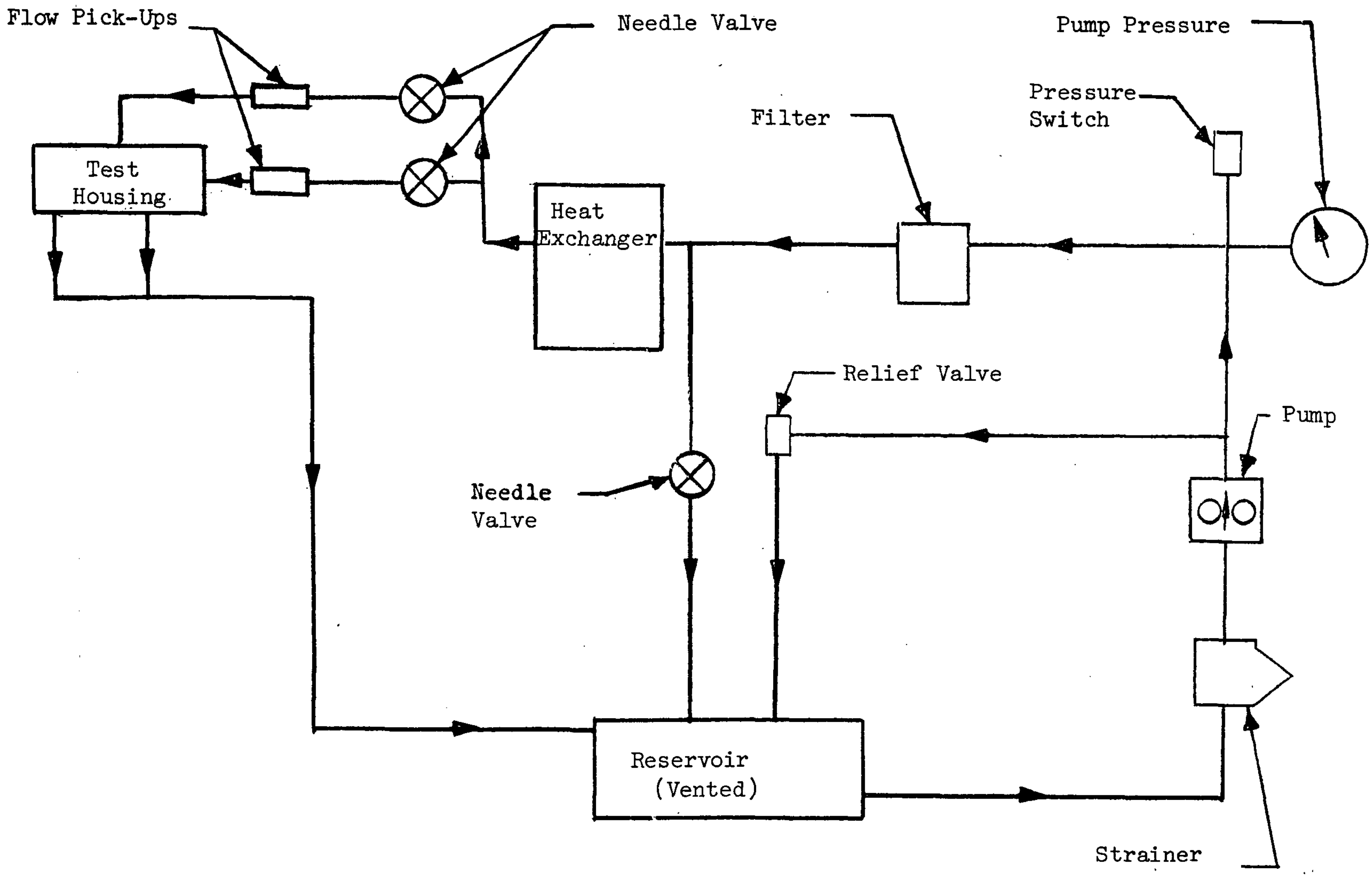

Figure 20. Lubrication System Schematic

Section III

Fabrication

The test rig and test seals were fabricated after the rig and seal designs were approved by the NASA Project Manager.

Figure 21 shows the components of the dual element split ring seal, consisting of a sleeve and carrier with rings installed in the carrier grooves. The components of the two piece ring are shown in Figure 22.

Figure 23 shows the components of the circumferential seal and Figure 24 shows the assembled seal.

The test rig is shown in Figure 25.

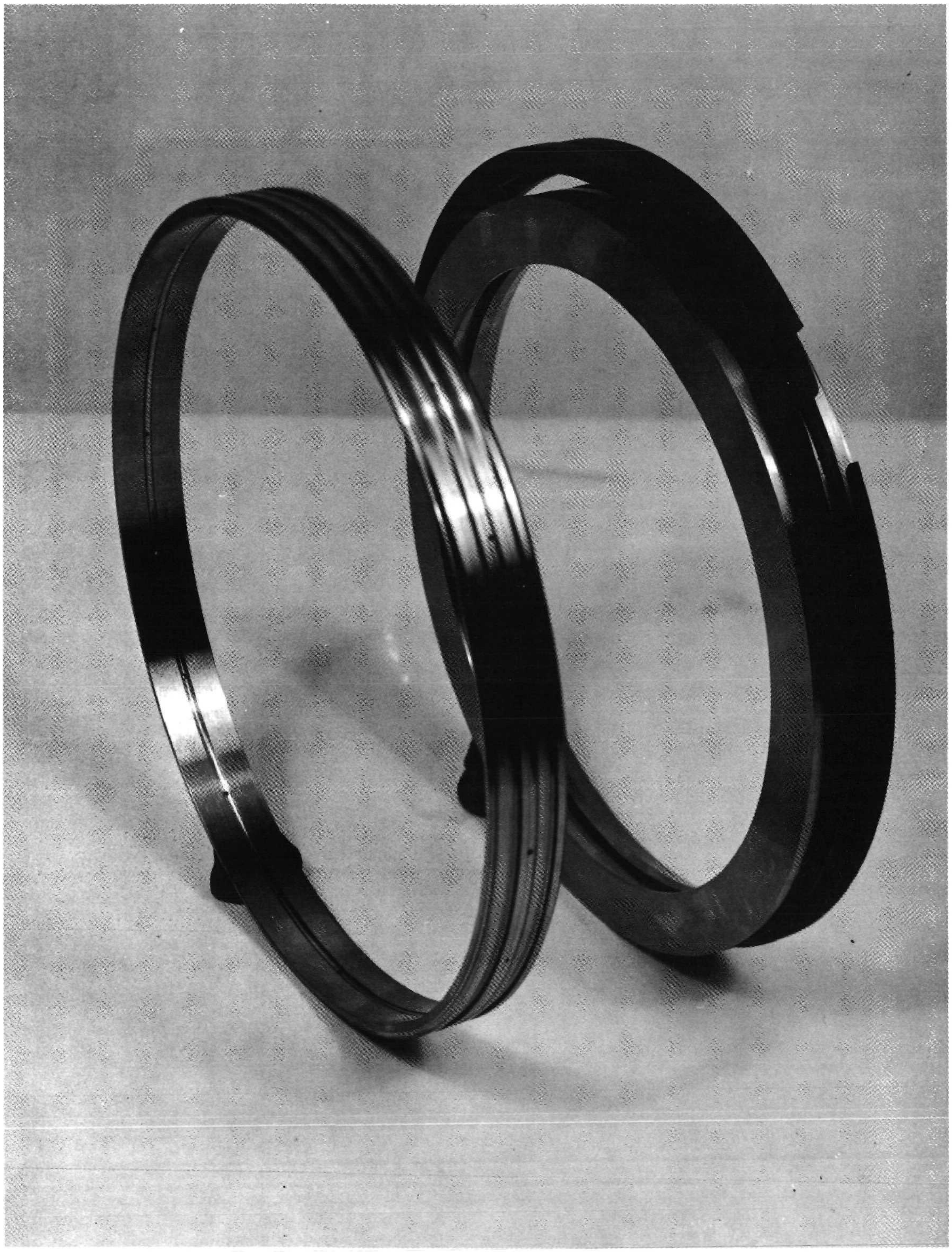

Figure 21. Components of Dual Element Split Ring Seal.

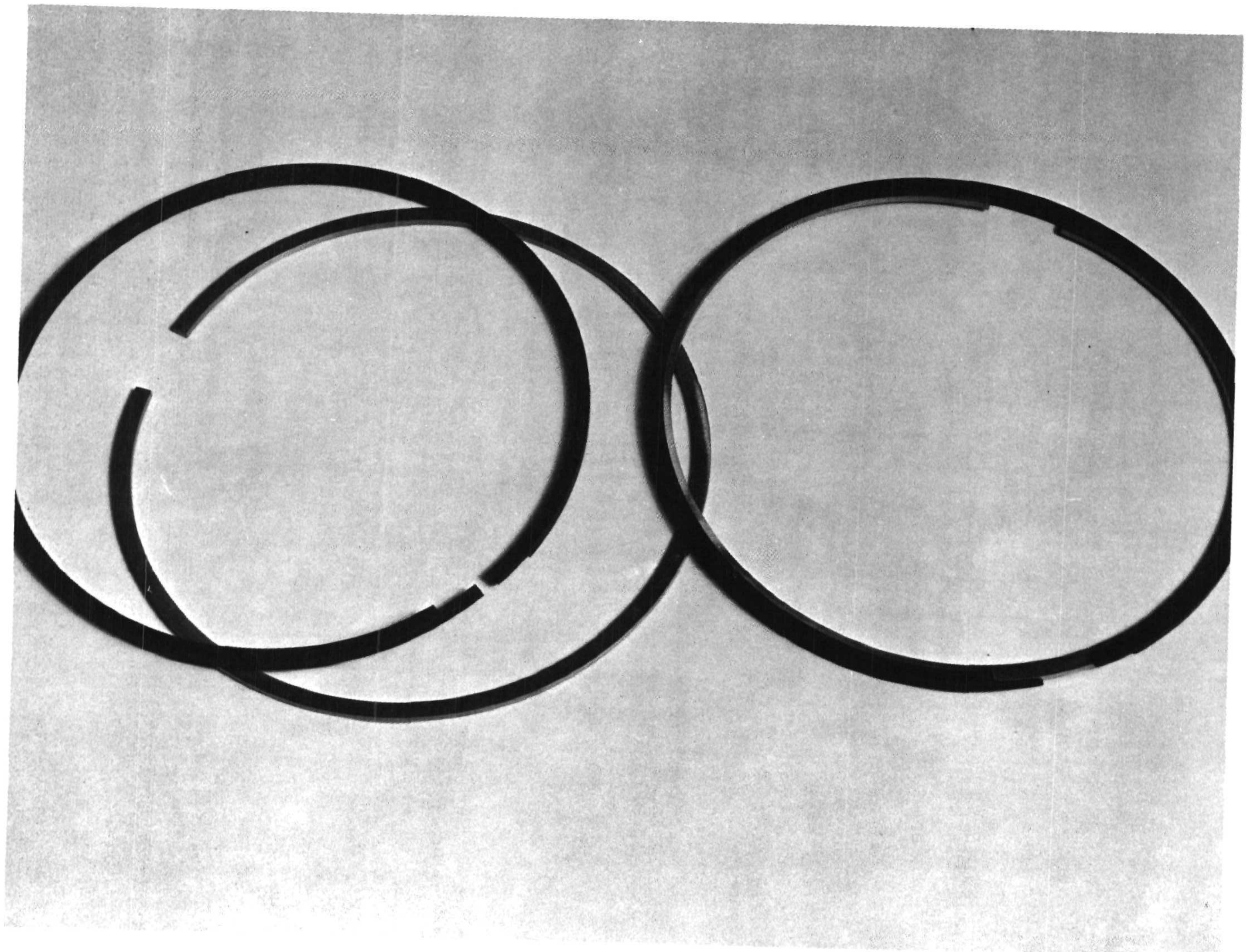

Figure 22. Two Piece Ring.

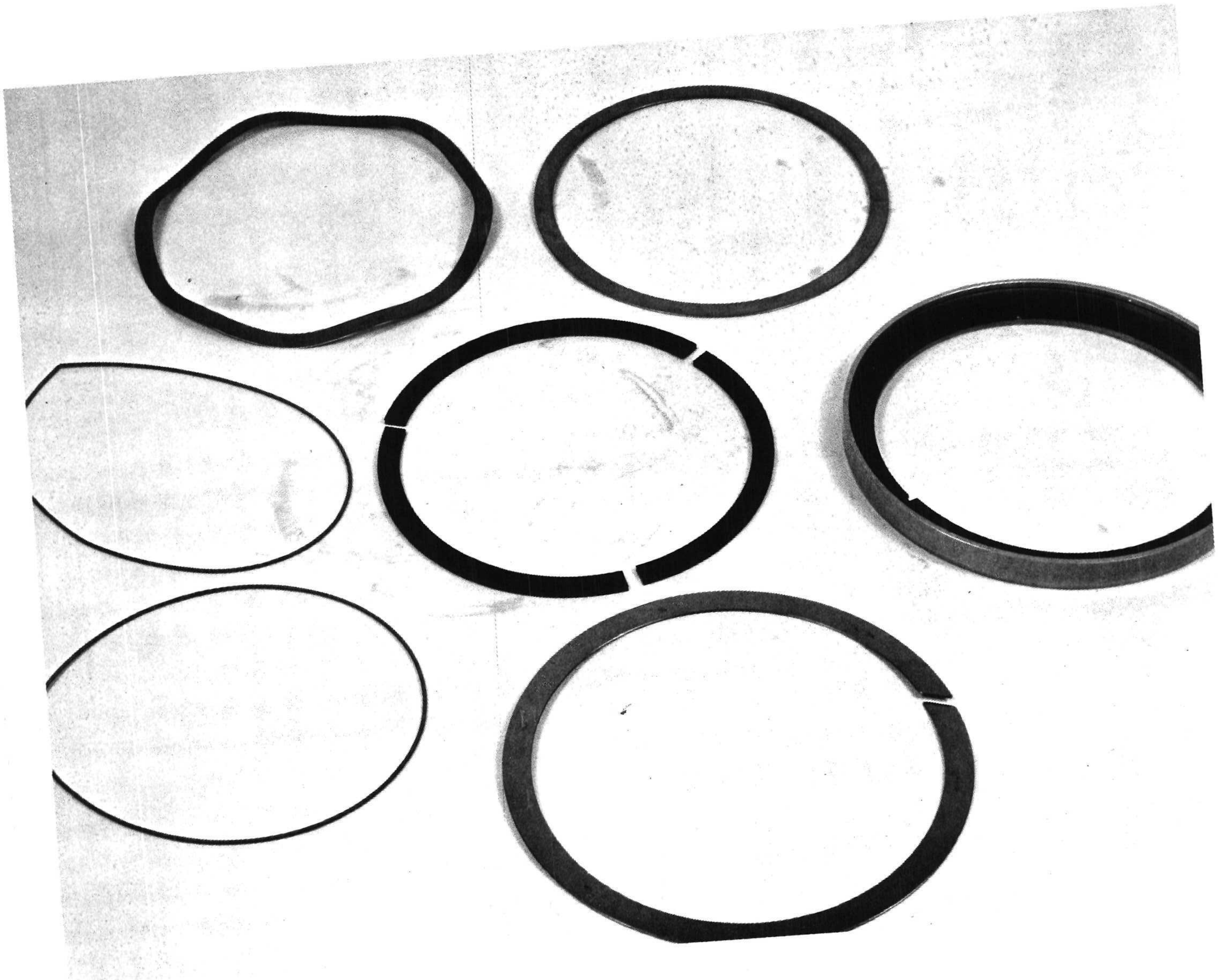

Figure 23. Components of Circumferential Seal.

Figure 24. Circumferential Seal.

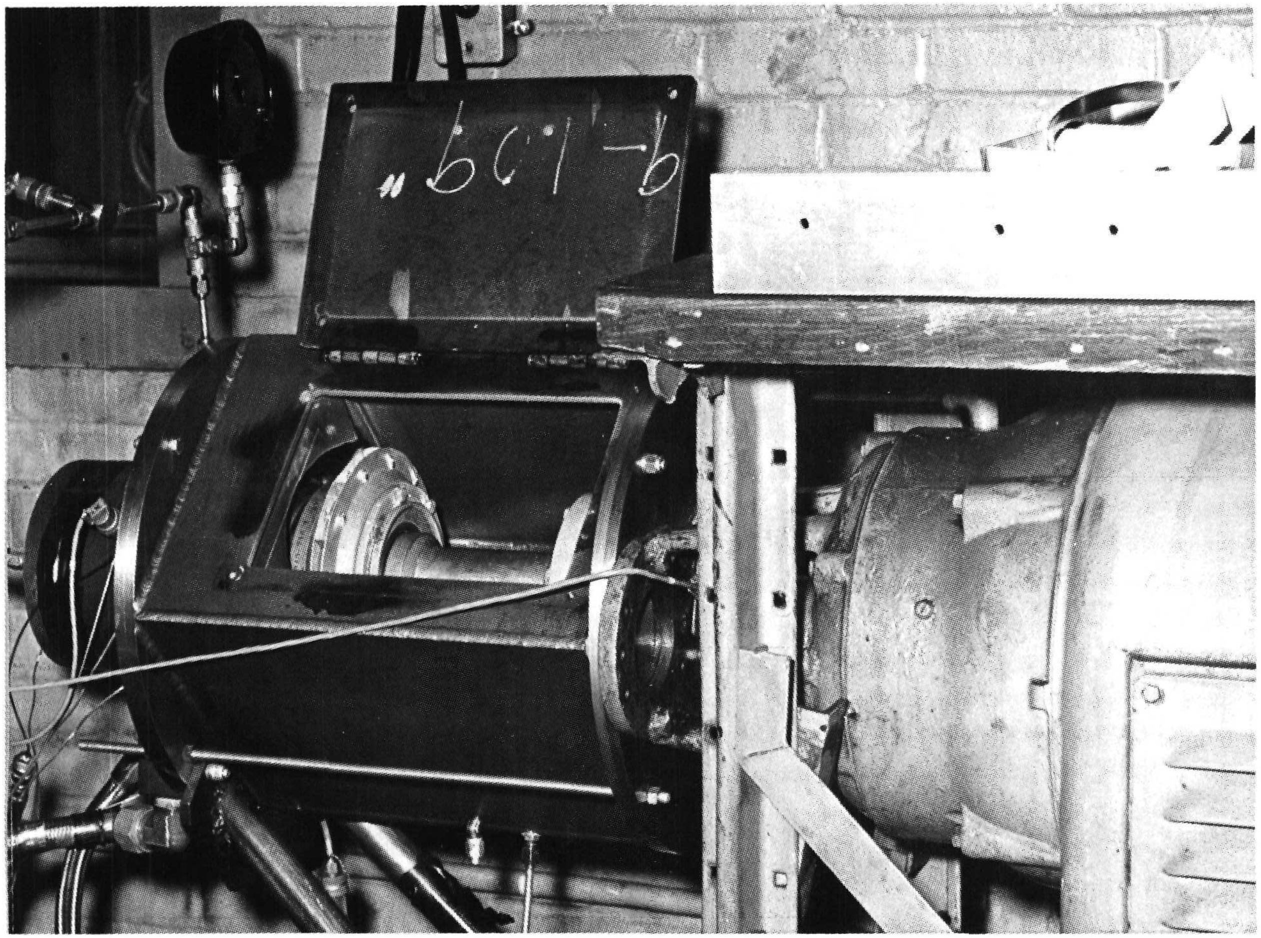

Figure 25. Seal Test Rig.

Section IV
Testing and Discussion of Results

During the test phase of the program both the dual element split ring seal and the circumferential seal were evaluated. The test of each seal was divided into two phases. The first phase was a screening test where the seal was reworked and refined until 25 hours of successful operation was obtained. When this was achieved, a second seal, with the refinements determined to be necessary during the screening test, was tested for 100-hours.

The lubricant temperature in the seal area, 380 - 389 K (225° - 240°F), and the shaft rotational speed, 6600 rpm, were held constant throughout the test (Stabilized lubricant temperatures could not be reached in some ring seal test, because of excessive seal leakage). The lubricant flow rates to the support bearing and the freewheel unit were varied during the test to determine the effect of lubricant flooding on seal performance. The testing and the results of each seal design will be discussed separately.

As shown in Table 3, the dual element split ring seal was reworked several times in an effort to obtain less than 1 c.c./hour of oil leakage. The reworked procedures were directed toward drainage in the seal area since it was known from previous experience that gross seal leakage would occur if flooding condition existed.

From results of the first screening test, it was evident that there was insufficient drainage for successful operation of the dual element split ring seal. As shown in Figure 26, the lowest point of the oil drain is at an angle of .582 radians (33.4 degrees) to the vertical and an oil head of approximately 13.45 m.m. (.53 inches) must be established before any drainage will occur. Locating the drain in this position is typical of helicopter transmission seal applications and is done to insure lubrication for lip seals. With the drain in this position, oil will at least cover the lower portion of the ring seal, as shown in Figure 27. The actual quantity of lubricant in the seal area in terms of lubricant in swirling motion and actual head height was not determined.

As shown in Table 3, the flooded seal ran .3 hours with a total leakage of 403 c.c. After disassembly the seal was inspected and found to be in excellent condition. The rings exhibited no wear or signs of heat discoloration. The cause of the high leakage rate was attributed to the drainage problem.

The following steps were taken to relieve the flooding condition:

1) Three 6.35 m.m. (.250 inch) drain holes were machined in the seal housing and connected directly to the oil reservoir. The holes were placed as shown in Figure 28. From Table 3 it can be seen that the leakage rate was substantially reduced but was still unsatisfactory.

2) The three drain lines were then connected to the pump inlet in an effort to pump the oil out of the seal area. This approach did not work because the pump tended to create a

TABLE 3 - Dual Element Split Ring Seal Test Data

Drainage Configuration	Temperature in Seal Area K (°F)	Pressure in Seal Area N/cm^2 (psi)	Lubricant Flow Rates c.c./sec. (GPM) Support Bearing	Freewheel Unit	Leakage c.c./hour
Original Design Configuration - 0.3 hours	326 to 329 (127 to 132)	0 (0)	27.1 (.43)	31.5 (.50)	1342
Modified Seal Housing - Three 6.38 m.m. (.25 inch) drain holes connected to reservoir -0.9 hours	357 to 363 (183 to 193)	.1375 to .413 (.2 to .6)	25.2 (.40)	27.75 (.44)	422
Modified Seal Housing - Three 6.35 m.m. (.25 inch) drain holes connected to pump inlet port - 1.0 hrs.	358 to 359 (185 to 187)	.344 (.5)	27.75 (.44)	30.25 (.48)	700
Scavenge Pump - Connected to the three 6.35 m.m. (.25 inch) drain holes - 0.7 hours	353 (176)	.344 (.5)	26.5 (.42)	32.8 (.52)	286
Scavenge Pump - Connected to original drain port and to three 6.35 m.m. (.25 inch) drain holes - 1.0 hour	368 to 372 (203 to 210)	0 (0)	35.3 (.56)	63 (1.0)	2
Modified Seal Housing with 6.35 x 54 m.m. (.25 x 2.125 inch) slot connected to reservoir - 4.0 hours	383 to 389 (230 to 240)	0 (0)	34.05 to 36.6 (.54 to .58)	34.05 (.54)	65
Modified Seal Housing with 6.35 x 54 m.m. (.25 x 2.125 inch) slot connected to reservoir					
1.5 hours	383 (230)	.689 to 1.24 (1.0 to 1.8)	35.2 (.56)	32.8 (.52)	26.6
.9 hours	389 (240)	.689 to 1.24 (1.0 to 1.8)	34.05 (.54)	25.2 (.4)	22.2
1.6 hours	382 to 383 (227 to 230)	.689 to .758 (1.0 to 1.1)	32.8 to 34.05 (.52 to .54)	18.85 (.3)	37.5
.5 hours	383 to 385 (230 to 233)	.619 to .689 (.9 to 1.0)	32.8 (.52)	12.6 (.2)	40
.5 hours	387 to 390 (236 to 241)	.619 to .689 (.9 to 1.0)	34.05 (.54)	6.3 (.1)	40
.5 hours	386 to 388 (235 to 239)	.619 to .689 (.9 to 1.0)	35.2 (.56)	3.14 (.05)	20
.3 hours	381 to 383 (225 to 228)	.619 to .689 (.9 to 1.0)	32.8 (.52)	0 (0)	18.3

Figure 26. Housing Drain Location.

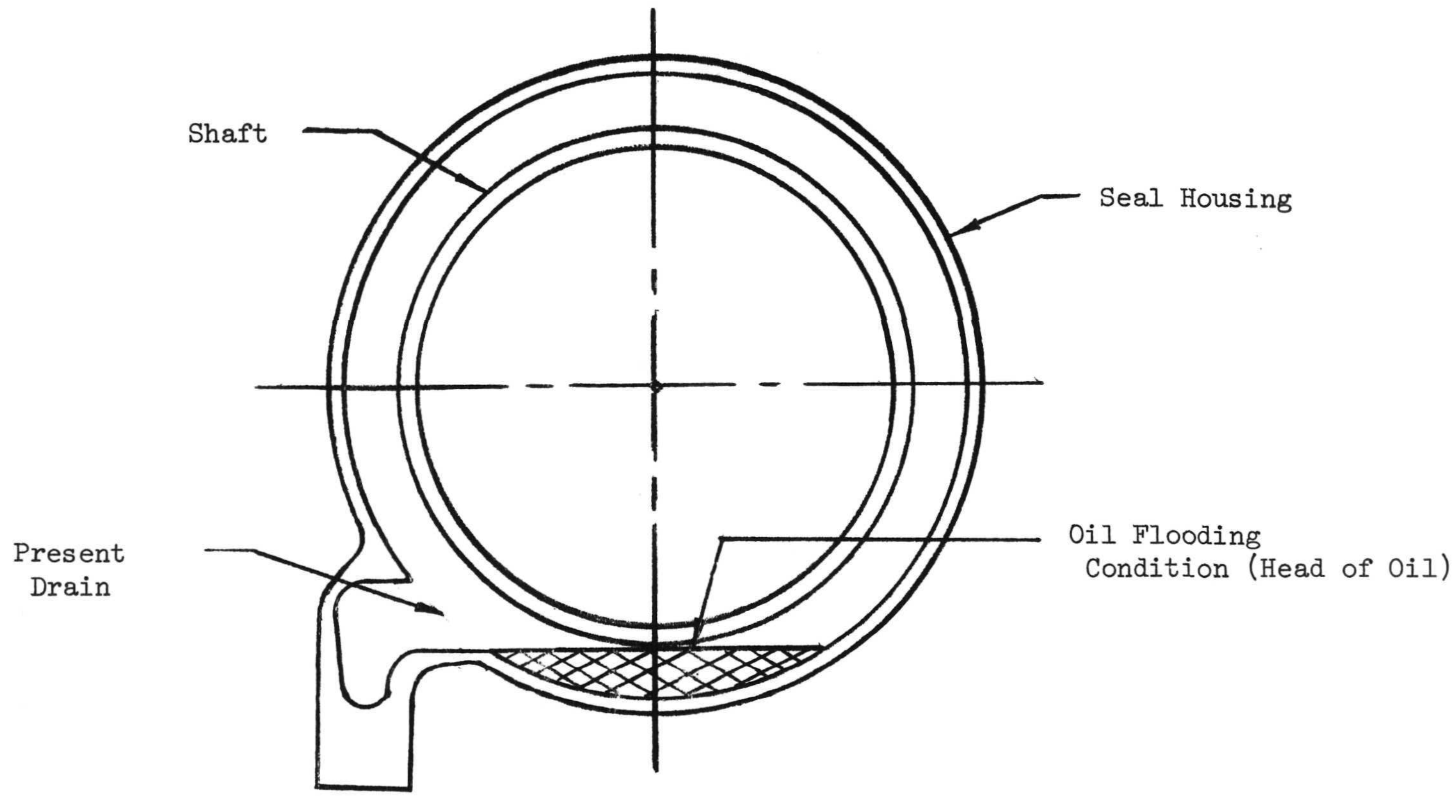

Figure 27. Flooding of Ring Seal.

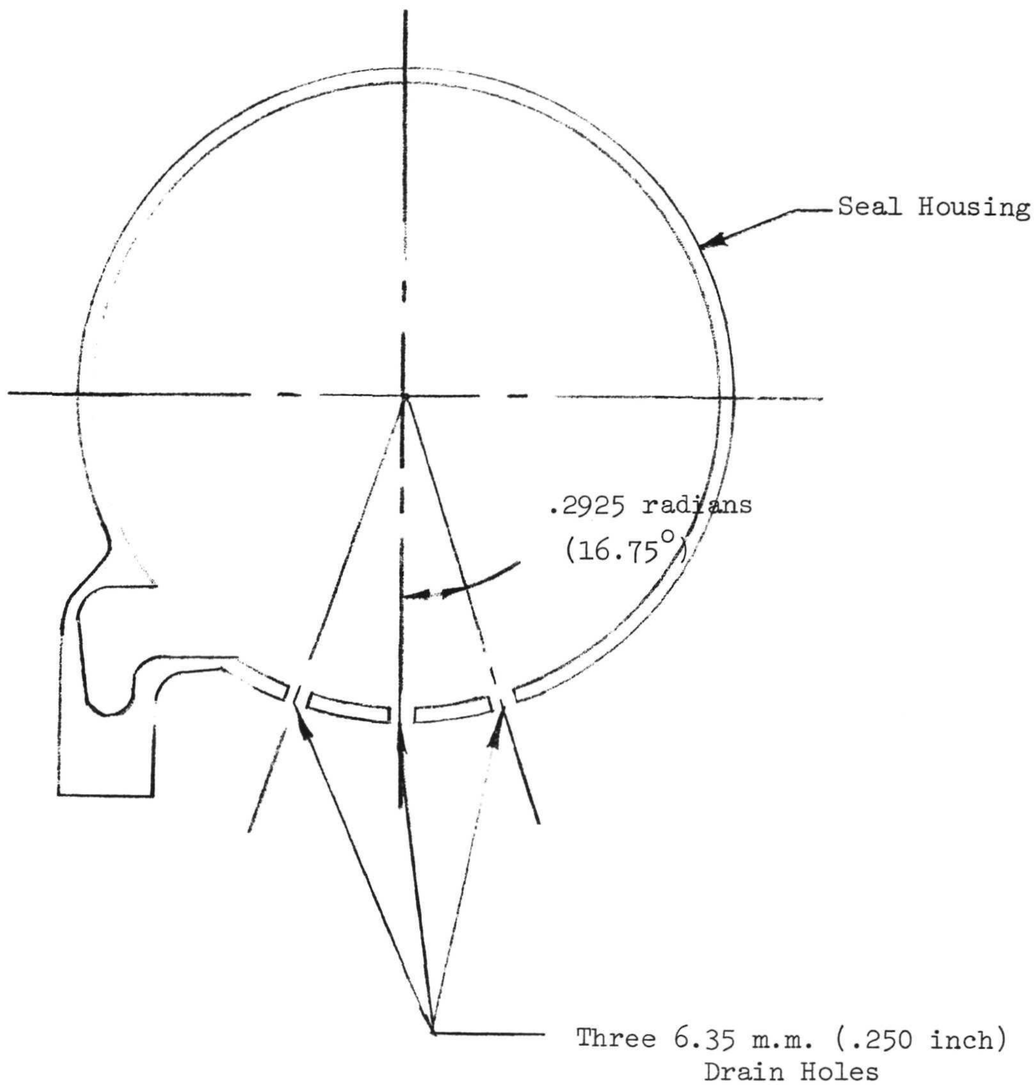

Figure 28. Modification of Seal Housing with Three 6.35 m.m. (.250 inch) Drain Holes.

back pressure and pump oil into the seal area.

3) An oil scavenge pump was connected to the three new drainage lines. The leakage was reduced to 286 c.c./hours, but was still unsatisfactory.

4) An oil scavenge pump was connected to the in-service drain port and to the three new drain lines. This configuration was successful and reduced oil leakage to 2 c.c./hour. Although this leakage is higher than the test allowable, it is considered acceptable in most transmission applications. It would be impractical to incorporate this approach into production gearboxes due to the added weight of an extra pump and drain lines.

5) The seal housing was further modified by machining a drainage slot 6.35 m.m. (.250 inch) wide by 54 m.m. (2.125 inch) long at the bottom of the seal housing as shown in Figure 29. An aluminum manifold was bonded to the outside diameter of the seal case and connected to the reservoir. Increased drainage capacity was established with this approach. The leakage rate recorded was 65 c.c./hour which was still unsatisfactory.

   This configuration was also tested by reducing the flow to the freewheel unit (Table 3) while the flow to the support bearings remained constant. Although leakage decreased, it was still unsatisfactory.

It was concluded that the dual element split ring could not be effective without major modifications to the seal housing and oil drainage system. Testing of the split ring seal was suspended and testing of the circumferential seal was initiated.

The screening test of the circumferential seal was completed after one rework. During the initial screening test the seal leaked 70 c.c. in 5.75 hours (12.2 c.c./hour). An oil flow of 37.9 c.c./sec. (0.6 gal/min.) was supplied to both the support bearings and the freewheel unit. Inspection of the seal and the seal runner revealed no rubbing contact. This lack of contact is due to the pressure of a lubricant film between the carbon element and the shaft. To obtain boundary lubrication and thus a smaller leakage gap, the garter spring load was increased from .92 N/cm (.0521 lb/in) of circumference to 1.84 N/cm (.1042 lb/inch) of circumference.

The modified seal was installed and the oil flow to the freewheel unit was increased to 67.0 c.c./sec. (1.06 gal./min.). As shown in Table 4 and in Figures 30 and 31, the seal completed the 25-hour screening test with an average leakage rate of .74 c.c./hour. No wear of the carbon graphite rings and no distress or measurable wear of the seal runner was evident (Figure 32).

A second circumferential seal with 1.84 N/cm (.1042 lb/inch) springs was installed in the test rig. The flow rate to the support and the freewheel unit

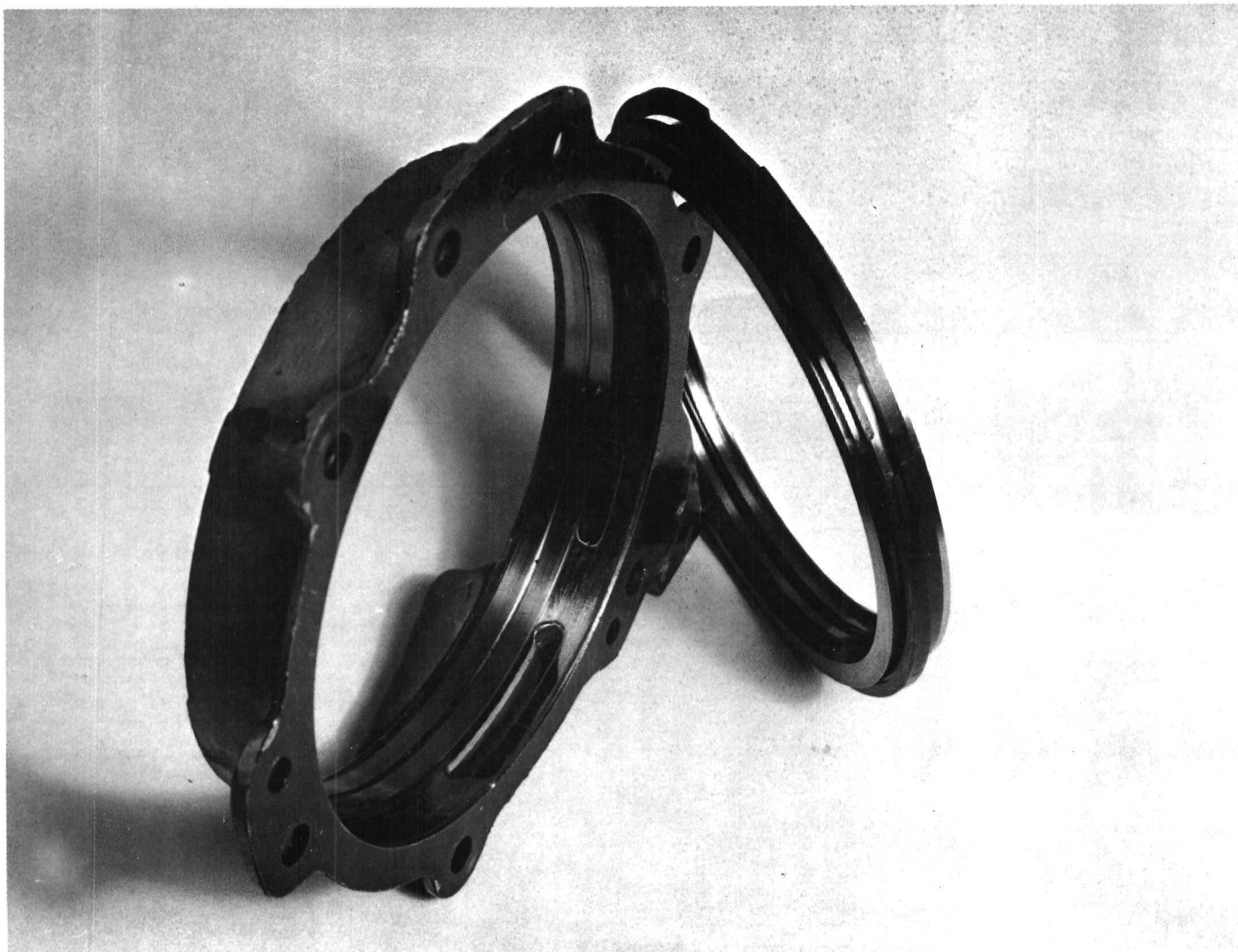

Figure 29. Modified Seal Housing with Rectangular Slot.

TABLE 4

MEAN TEST DATA FOR CIRCUMFERENTIAL SEAL

	25 Hour Test	100 Hour Test
Temperature - Degrees K	384 (232°F)	384 (232°F)
Oil Pressure - N/cm^2 in Seal Area	$\sim$0.0	$\sim$0.0
Oil Flow Rate - Cm^3/Sec.		
Freewheel Feed	67.0 (1.06 GPM)	79.5 (1.26 GPM)
Stack Bearing Feed	37.9 (0.60 GPM)	41.0 (0.65 GPM)
Shaft Rotational Speed - RPM	6600	6600
T.I.R. Radial Runout - Hand Turn - m.m.	.0508 (.002 inch)	.0508 (.002 inch)
Average Seal Leakage Rate - cm^3/Hr.	0.74	0.56

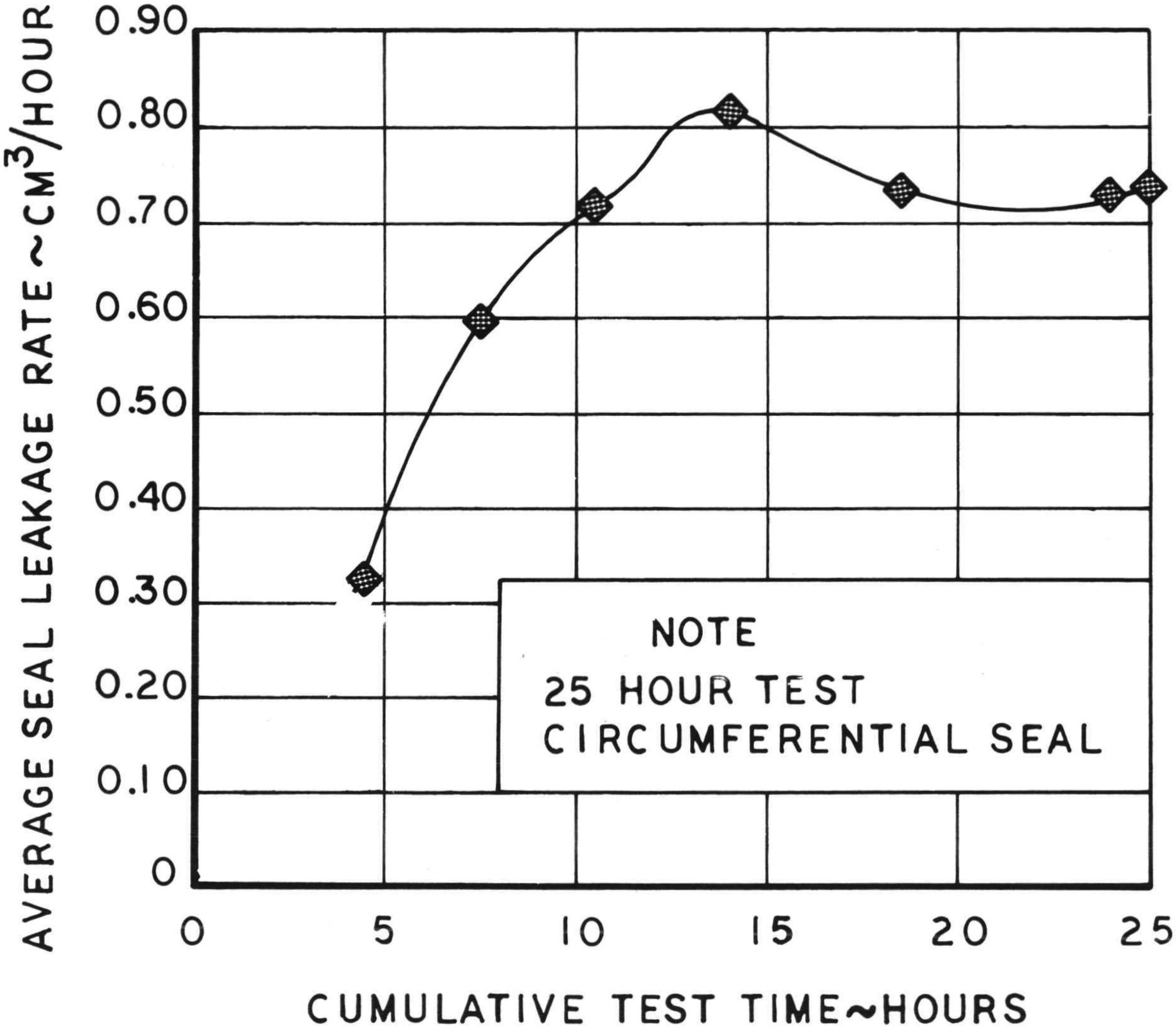

Figure 30. Leakage Rate for 25 Hour Circumferential Seal Test

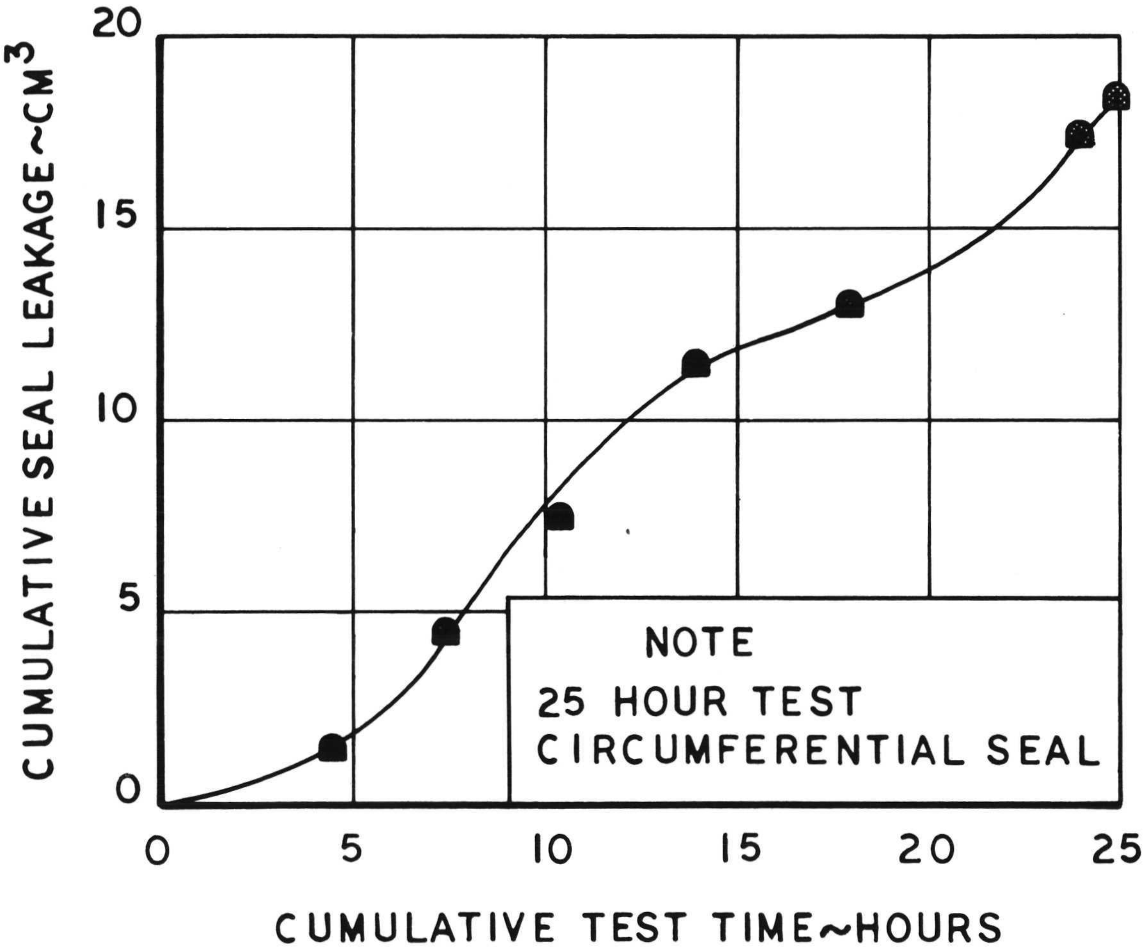

Figure 31. Cumulative Leakage for 25 Hour Circumferential Seal Test

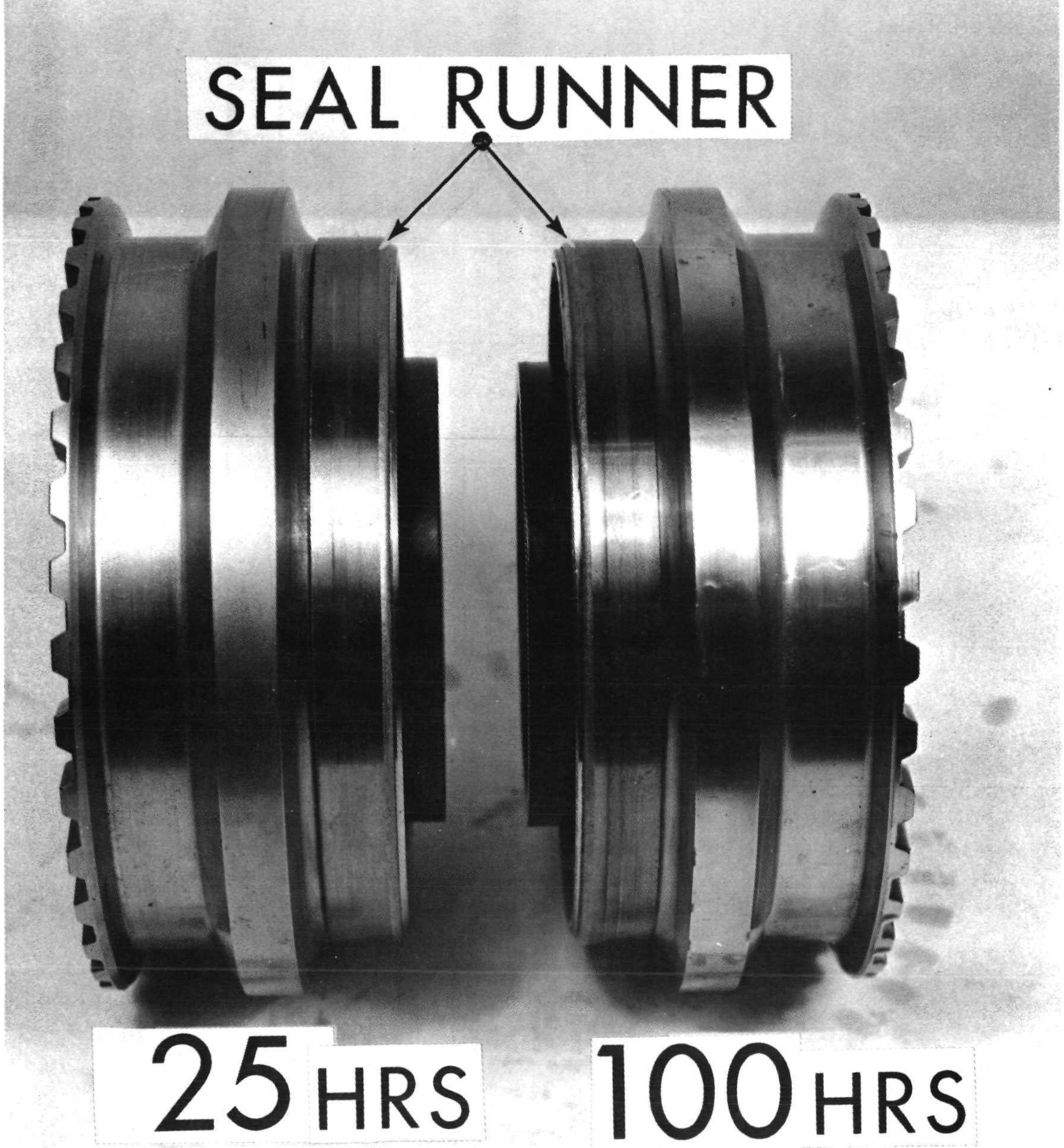

Figure 32. Circumferential Seal Runner Used in 25-Hour and 100-Hour Test.

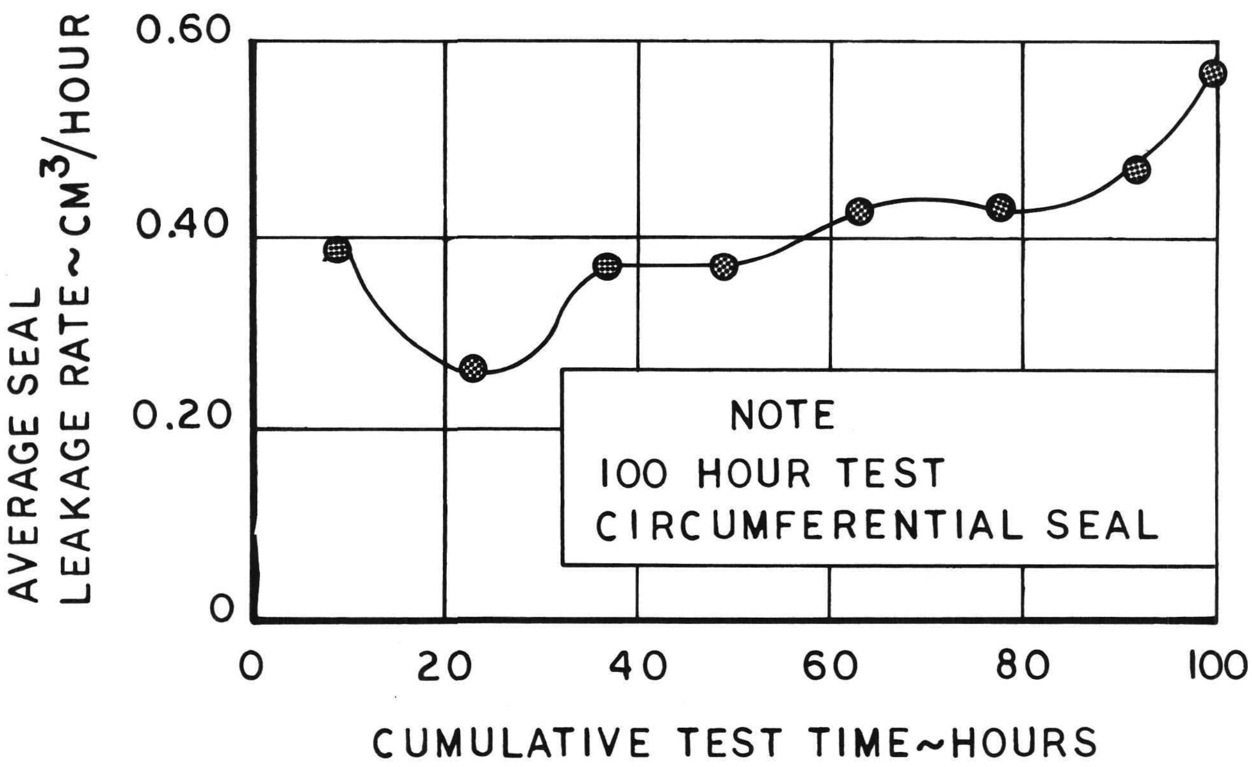

Figure 33. Leakage Rate for 100 Hour Circumferential Seal Test.

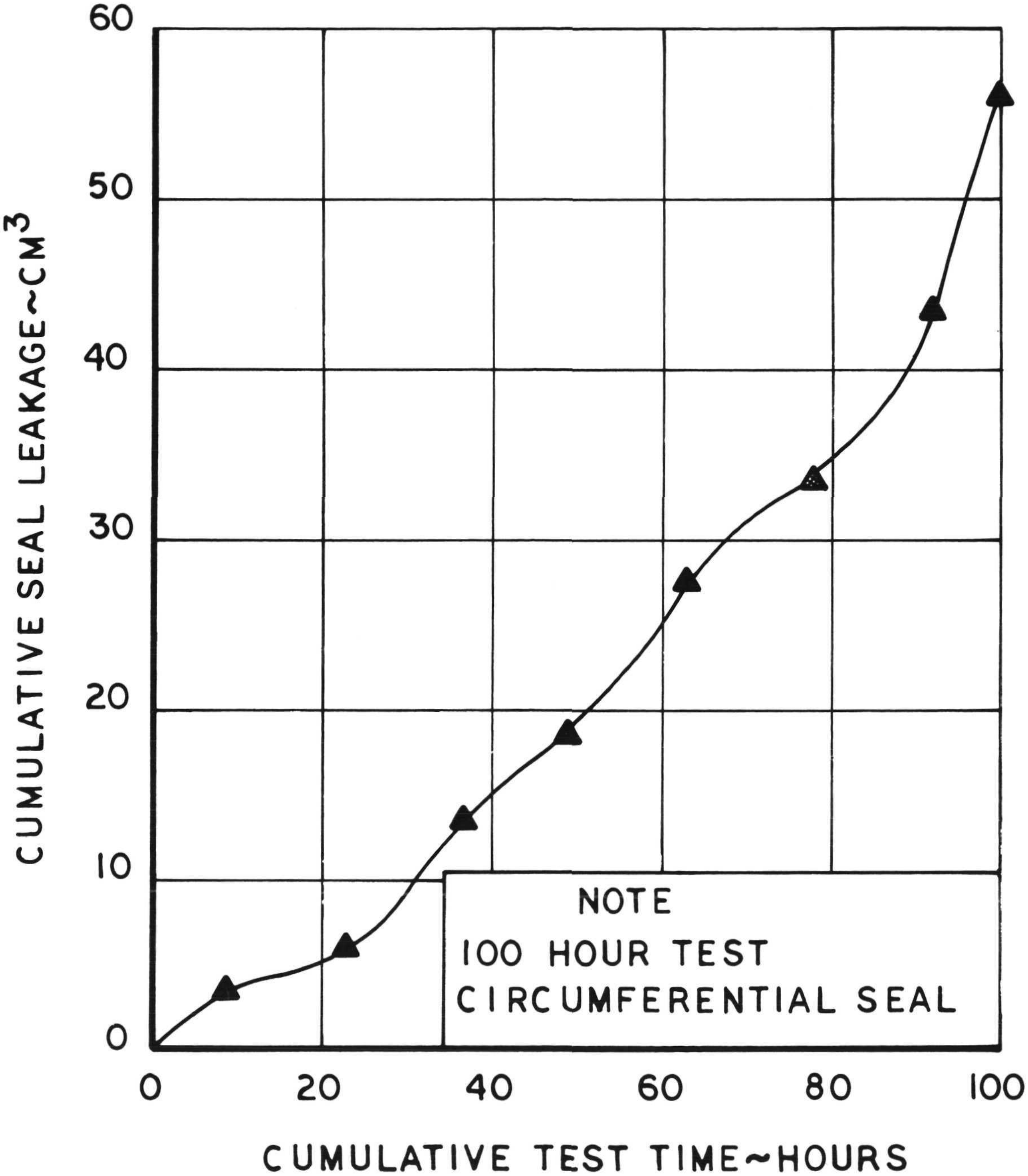

Figure 34. Cumulative Leakage for 100 Hour Circumferential Seal Test.

was increased to 41.0 c.c./sec. (.65 gal./min.) and 79.5 c.c./sec. (1.26 gal. gal./min.). The seal was tested for 100 hours with approximately 40 stop-start cycles. The data obtained from this test is shown in Table 4 and in Figures 33 and 34.

The primary ring and backer ring radial wall thickness were measured prior to testing. Measurement after the test showed a reduction in thickness of 0.0 m.m. to .0127 m.m. (0.0 to .0005 inch) depending on the angular location of the measurement. The reduction in wall thickness probably occurred during initial running and should present no problems over extended operating periods. Slight distress to the seal runner was observed although it was less than 2.54 m.m. (.0001 inch).

An attempt to simulate the auto-rotative mode of the transmission by reversing the direction of rotation and overrunning the freewheel unit, was not successful because the angle cut of the primary element gap (Figure 17) tended to pump oil to the air side of the seal. During auto-rotative maneuvers of the aircraft, this would not occur since the shaft always rotates in the same direction.

A 40-hour environment test was then conducted on the successful 25-hour seal under the following conditions:

- 100% shaft speed for 55 minutes and 110% shaft speed for 5 minutes.
- seal subjected to 2 c.c. of 140 mesh silica flour (simulating Southeast Asia dust) every hour.
- support bearing oil flow of 39.9 c.c./sec. (.6 gal./min.).
- freewheel unit oil flow of 82.1 c.c./sec. (1.3 gal./min.) for 24 hours and 126 c.c./sec. (2.0 gal./min.)for the last 16 hours.
- a stop-start cycle every hour.

A leakage rate of .124 c.c./hour was recorded for the first 24-hour period and .095 c.c./hr. for the final 16-hour period. Due to abrasive action of the silica flour a wear groove was produced on the seal runner about .071 m.m. (.0028 inches) during the 40 hour test. Considering the severity of the test this was considered satisfactory performance, especially since the leakage actually decreased during the run. The carbon-graphite rings wear was approximately .0305 mm (.0012 inches). The profile trace of the seal runner is shown in Figure 35. Figure 36 and 37 shows the seal and the seal runner respectively.

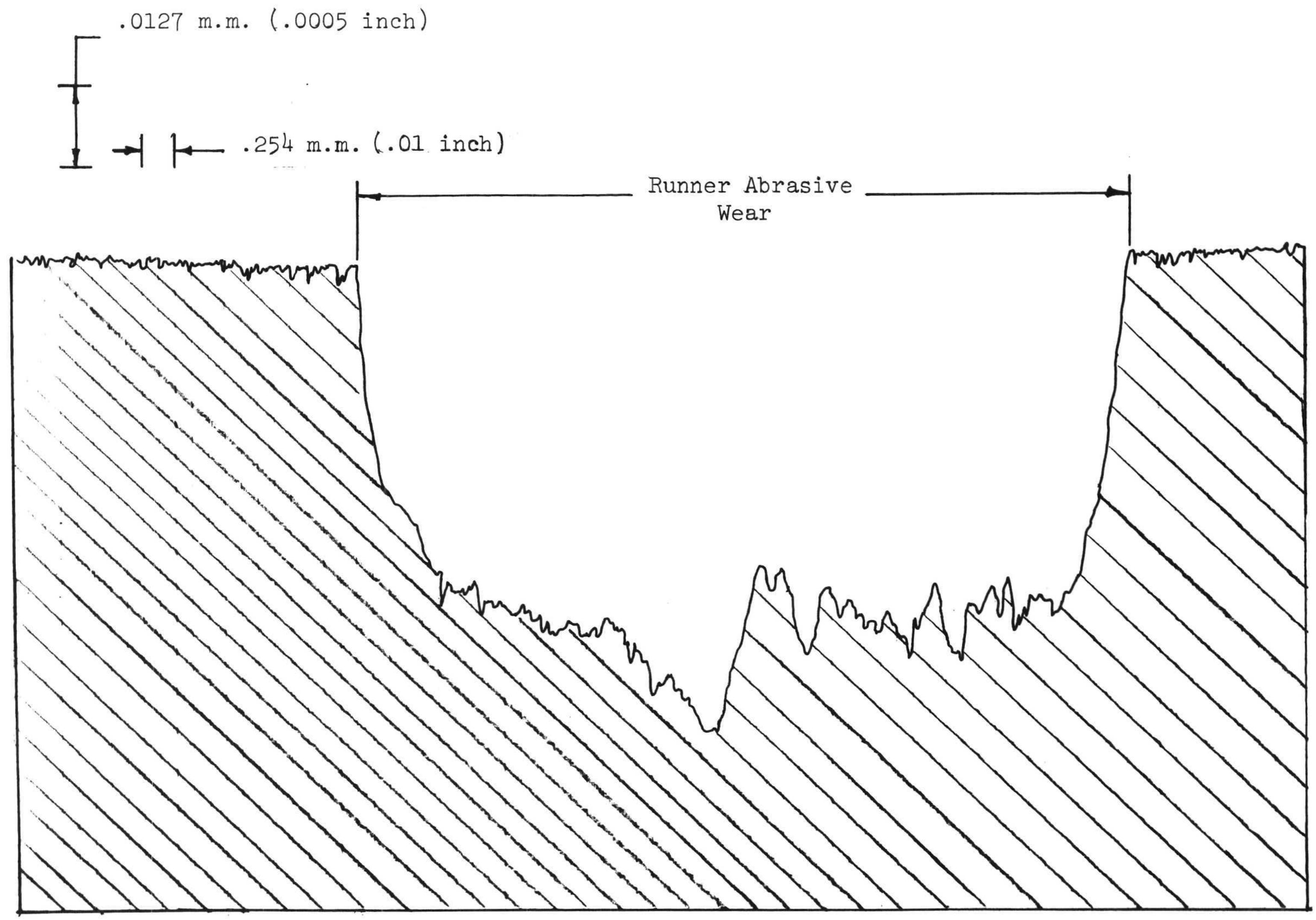

Figure 35. Profile Trace of Seal Runner from 40-Hour Environmental Test.

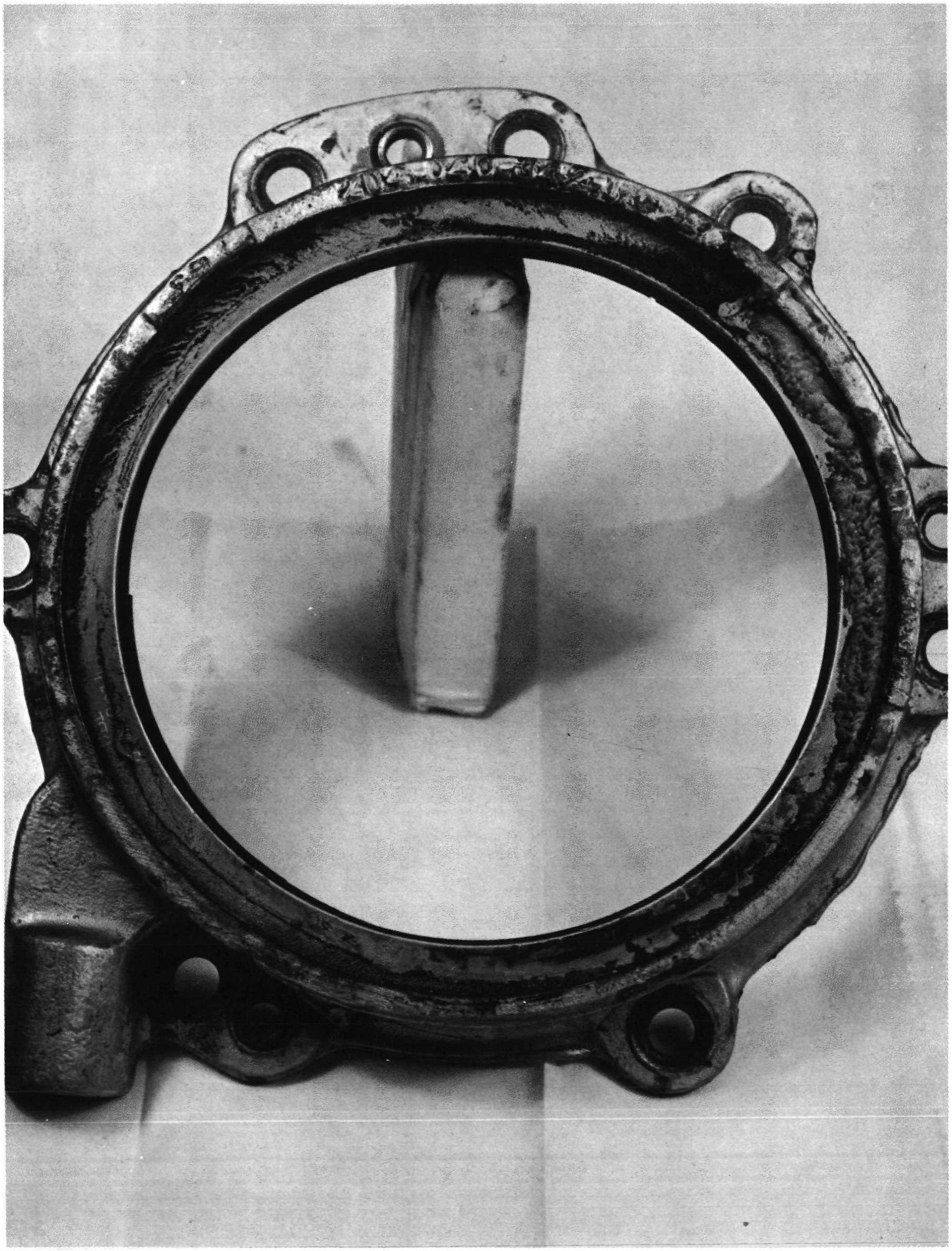

Figure 36. Circumferential Seal from 40-Hour Environmental Test

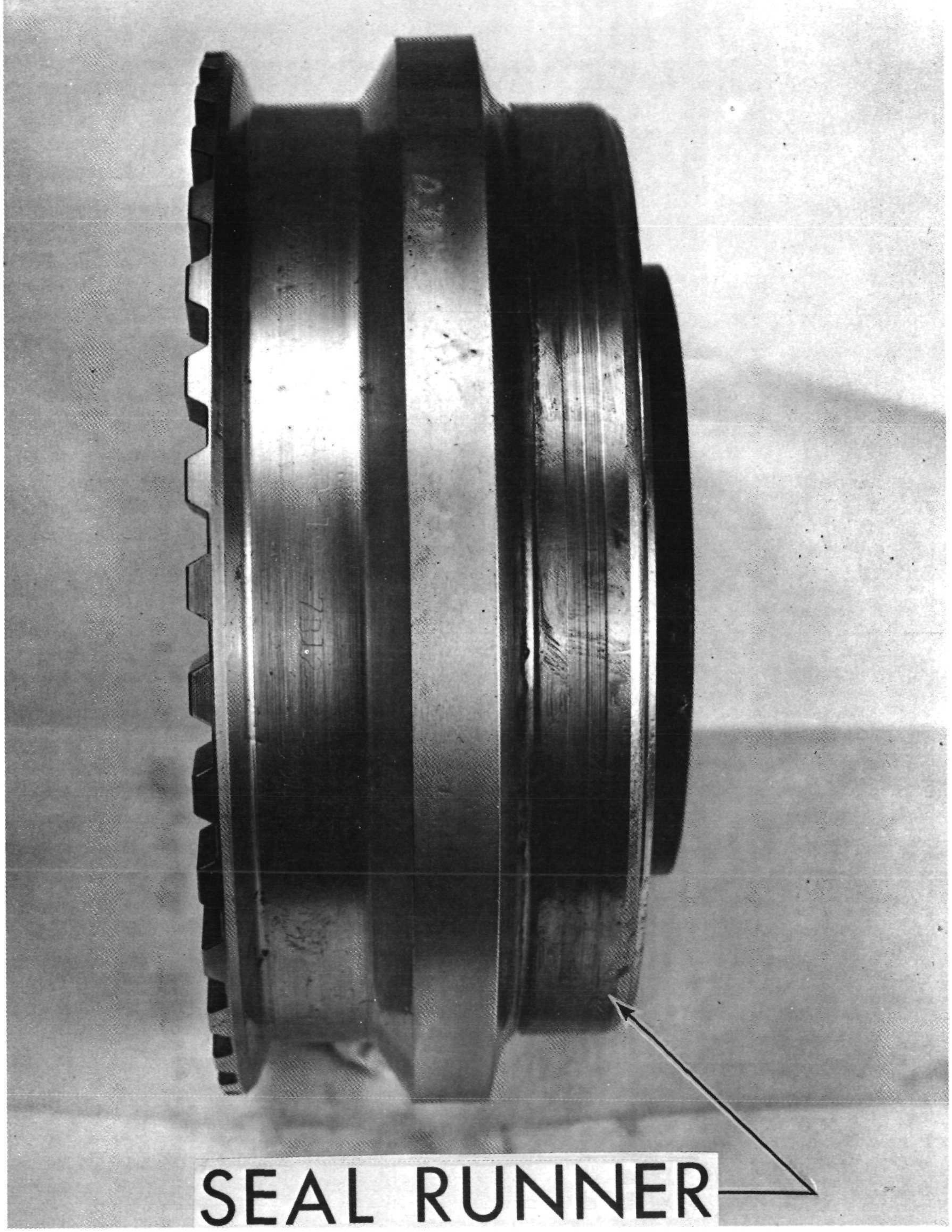

Figure 37. Seal Runner from 40-Hour Environmental Test, View A.

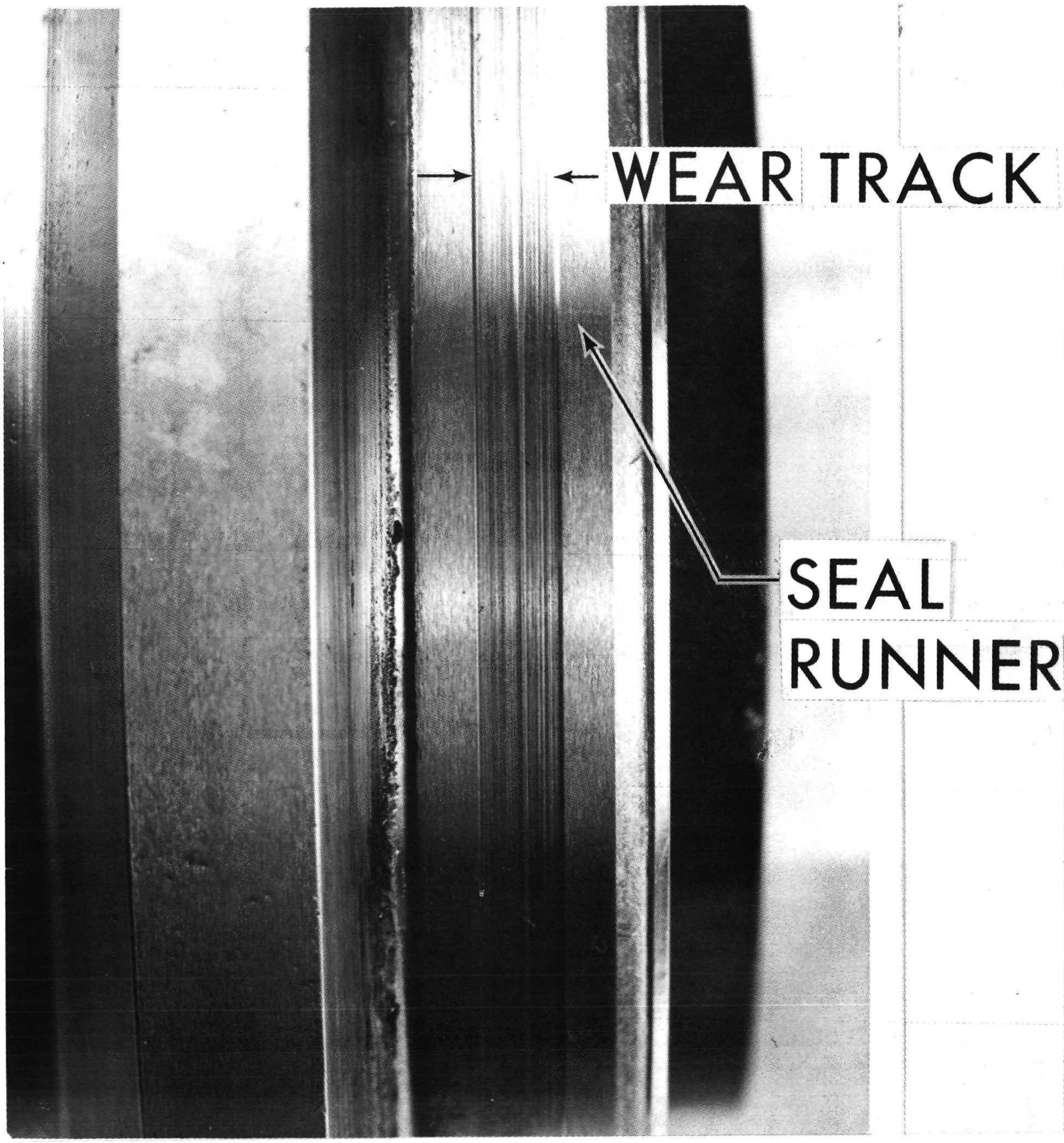

Figure 37. Seal Runner from 40-Hour Environmental Test, View B.

Section V

CONCLUSIONS

The results of the tests performed under Task II provide considerable information about the feasibility of using positive contact seals (circumferential seals) and controlled clearance seals (dual element split ring seals) in high speed helicopter transmission applications. Specific conclusions reached as a result of the testing on seals for a 136 mm (5.355 inch) diameter shaft at 6600 rpm (9260 fpm) at a 384 K (230 $^{\circ}$F) lubricant temperature and at a shaft radial runout of .0508 mm (.002 inch) are listed below:

- Circumferential seals operated with leakage rates less than 1 c.c./hour in a transmission environment of splash lubrication and partial flooding with near zero pressure differential. This leakage is within U.S. Army specifications.

- Wear of the carbon-graphite elements in circumferential seals was minimal and is not a major consideration.

- Abrasive wear of the circumferential seal runner was caused by the introduction of silica flour. The amount of wear was not excessive considering the severity of the test.

- The projected wear life of the circumferential seal, based on the test data, is in excess of 1100 hours dictated by the performance criteria. From experience with other similar applications it is expected that the seal is capable of a number of 1100 hour tours of duty if proper attention is applied to assembly and disassembly procedures.

- The dual element split ring will leak excessively if exposed to a flooded condition and the leakage rate of the dual element split ring seal is directly related to the configuration and capacity of the oil drain system.

- The dual element split ring seal should only be used when excellent drainage is available.

Reference

1. "An Investigation of the Operation and Failure of Mechanical Face Seals; F. R. Orcutt; 4th International Conference on Fluid Sealing, Philadelphia, Pennsylvania, 1969. BHRA, Cranfield, Bedford, England.

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